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U. S. S. *ST. LOUIS*.—DESCRIPTION AND OFFICIAL TRIAL.

BY JAMES NELSON ALEXANDER, ASSOCIATE.

The *St. Louis* (Protected Cruiser No. 20) is a twin-screw cruiser of 9,700 tons displacement, built by the Neafie and Levy Ship and Engine Building Company, of Philadelphia, and is one of the three vessels of this class authorized by an Act of Congress, approved June 7, 1900, the others being the *Charleston* and *Milwaukee*. The contract for the *St. Louis* was signed March 11, 1901, the price being $2,740,000, exclusive of armor, armament, equipment, etc., which are supplied by the Government, and the time allowed for completion thirty-six months from date of contract. This time was extended to March 11, 1905, a second extension to March 11, 1906, and a third extension to July 11, 1906. The guaranteed speed was 22 knots per hour to be maintained successfully for four consecutive hours, on a mean draught of 22 feet 6 inches, corresponding to a displacement of 9,700 tons, and the air pressure in the ash pits not to exceed an average of

one inch water. If the speed fell below 22 knots, the contract inflicts a penalty at the rate of $50,000 per quarter knot for speeds between 22 and 21½ knots, and $100,000 per quarter knot for speeds between 21½ and 21 knots. In case of failure to reach a speed of 21 knots, it was optional with the Secretary of the Navy to reject the vessel or accept her at a reduced price and on conditions to be agreed upon. The first keel plate was laid July 31, 1902, and the ship was launched May 6, 1905.

HULL DATA.

Length between perpendiculars, feet and inches	424–00
on load water line (22 feet 6 inches), feet and inches	424–00
over all, feet and inches	426–06
of straight keel, feet and inches	340–05¼
Beam, extreme, feet and inches	66–00
at L.W.L. (22 feet 6 inches), feet and inches	66–00
molded, feet and inches	65–01¼
Depth, molded, main deck side at M.S., feet and inches	40–02
upper deck side at M.S., feet and inches	47–07⅜
Draught, normal, mean, feet and inches	22–06
Displacement, mean load (empty bunkers), tons	9,127.6
normal (22 feet 6 inches), tons S.W	9,663.7
full load (24 feet 7 inches), tons S.W	10,776.5
per inch at L.W.L. (22 feet 6 inches), tons	44.92
Area of midship section to L.W.L., square feet	1,294.4
L.W.L. plane (22 feet 6 inches), square feet	18,872.69
Wetted surface, square feet	31,838.0
Center of gravity of L.W.L. at M.S., feet	2.14
bouyancy above bottom of keel, feet and inches	12–10¼
forward of M.S., feet	1–03
Transverse metacenter above center of bouyancy, ft. and ins	14–00⅜
Longitudinal metacenter above center of bouyancy, ft. and ins	509–08⅜
Coefficient of fineness, block	.53
of M.S	.87
L.W.L	.67
cylindrical	.61
Number of frames	114
watertight compartments	186

The *upper deck* extends from frame 27 forward to frame 87 aft. A day cabin for the commanding officer is built on this deck under the forward bridge, and above it on the bridge is the chart house. All boats, except two whale boats which swing in davits, are stowed on this deck when at sea.

Type.		Location.	Weight with crew.
Steam cutter,	36-foot,	Port,	16,961
	33-foot,	Starboard,	14,436
Sailing launch,	36-foot,	Port,	9,954
	33-foot,	Starboard,	8,471
Cutter,	30-foot,	Starboard,	5,625
	30-foot,	Port,	5,625
	26-foot,	Starboard,	4,447
	26-foot,	Port,	4,447
	26-foot,	Starboard,	4,447
	26-foot,	Port,	4,447
	26-foot,	Starboard,	4,447
	26-foot,	Port,	4,447
Whale boat,	30-foot,	Starboard,	4,802
	30-foot,	Port,	4,802
Gig whale boat,	30-foot,	Starboard,	4,471
Dinghy,	20-foot,	Starboard,	2,080
	20-foot,	Port,	2,088
	14-foot,	Starboard,	1,325
Punt,	14-foot,	Starboard,	(about) 700
	14-foot,	Port,	(about) 700

Four electrically-operated boat cranes are located on this deck, their heels being stepped on the gun deck. The forward pair are located at frame space 47-48, port and starboard, and the after pair at frame space 66-67, port and starboard.

Four electrically-operated deck winches for coaling and general use are located in pairs, port and starboard, at frame space 44-45 and 67-68. Twelve 3-pounder semi-automatic guns are mounted on this deck, in pairs, port and starboard, at frames 28½, 31, 45, 69, 81½ and 85½; two 1-pounder R.F. guns, port and starboard, at frame 36; and two Colt automatic .30-caliber guns, port and starboard, at frame 78½.

A 1,200-gallon gravity tank is located forward on this deck and two 400-gallon tanks aft.

Bridges.—The *forward bridge* is triangular in shape, the forward end being rounded off symmetrically with the for-

ward end of the upper deck. Chart and pilot houses are located on this bridge, also two Gatling and two Colt automatic guns.

The *flying bridge* is just above the forward bridge. The extension forward of the foremast carries the steering wheel, helm and revolution indicators, engine telegraphs, whistle and siren pulls, etc. A searchlight is located on each wing of this bridge.

The *after bridge* is above the after end of the upper deck. On this bridge is located a wireless telegraph house, on top of which are two searchlights, and four Colt automatic guns.

Masts.—There are two steel military masts, the foremast at frame 31, and the mainmast at frame 80, each fitted with an upper and lower military top, a searchlight platform, spar topmast and signal yard.

HEIGHTS ABOVE DESIGNED L.W.L., 22 FEET 6 INCHES FORWARD AND AFT, WHICH IS 22 FEET 4$\frac{1}{8}$ INCHES ABOVE THE BASE LINE.

Masts:	
Truck light, top of, fore and main, feet and inches	131-01$\frac{1}{2}$
Upper top, floor, foremast, feet and inches	67-03$\frac{1}{2}$
Lower top, floor, foremast, feet and inches	56-03$\frac{1}{2}$
Upper top, floor, mainmast, feet and inches	66-11$\frac{1}{2}$
Lower top, floor, mainmast, feet and inches	55-11$\frac{1}{2}$
Tops, depth, feet and inches	2-06
Searchlight platform, foremast, feet and inches	75-03$\frac{1}{2}$
mainmast, feet and inches	74-11$\frac{1}{2}$
Flying bridge at center at No. 34, feet and inches	41-10$\frac{1}{2}$
side, feet and inches	41-04$\frac{1}{2}$
above main deck, feet and inches	23-03$\frac{1}{2}$
Forward bridge at center at No. 34, feet and inches	34-05$\frac{1}{2}$
After bridge at center at No. 80, feet and inches	34-04$\frac{1}{2}$
side, feet and inches	33-11$\frac{1}{2}$
Top of flat armor deck, feet and inches	3-03$\frac{1}{2}$
main-deck planking at stem, feet and inches	21-01$\frac{1}{2}$
stern, feet and inches	20-00$\frac{1}{2}$
wrist plate at stem, feet and inches	24-10$\frac{1}{2}$
beading at stern, feet and inches	20-05$\frac{1}{2}$

The *main deck* is a continuous deck extending the whole length of ship. Forward on this deck are two anchor davits, four chain controllers, two riding bitts for anchor chain, and four deck openings for the hawse pipes for anchor chain.

The anchor windlass is located in a covered pocket between the main and gun decks, and rising above the main deck, the chains passing out through chain pipes in the forward side of this pocket.

Two electrically-operated deck winches for anchor work, boat hoisting, coaling and general use are located at frame space 26-27, port and starboard.

Within the superstructure on this deck is the berthing space for the crew and hammock nettings; general mess issuing room, forward of No. 1 boiler hatch; bakery, forward of No. 2 boiler hatch; officers' and crew's galleys, between No. 1 and 2 boiler hatch; general mess pantry, aft of No. 3 boiler hatch; printing office, starboard side, opposite No. 4 boiler hatch; ordnance stores, port side, opposite No. 4 boiler hatch; paymaster's office, forward of engine-room hatch; executive officer's office, after compartment, starboard side; navigator's office, after compartment, port side.

One electrically-operated winch for general service is located at frame space 97-98 amidship.

Six 6-inch and six 3-inch R.F. guns are located on this deck.

Outside the superstructure the main deck is covered with 3½ inch by 3½ inch Georgia pine, inside the superstructure with linoleum.

Boat davits are fitted for two 30-foot cutters forward of the superstructure, two 20-foot dinghies abreast of the forward end of superstructure, and for two 30-foot whale boats abreast of the after end of superstructure, in pairs, port and starboard. Six 6-inch and six 3-inch guns are located on this deck.

On the *gun deck* beginning forward are arranged: Crew's showers; crew's water closets; windlass engine and windlass compartment; berthing space for crew; general mess bread room; laundry, starboard; commissary's stores, port; upper casemate, armored; distiller room, aft No. 3 boiler hatch; armory, forward side of engine-room hatch; officers' lavatories, port; wardroom officers' rooms, five starboard, two port; ordnance officer's office, starboard; wardroom pantry,

aft of engine-room hatch; captain's office, pantry, cabins (main and after); staterooms (two) and baths (two).

There are eight 6-inch guns, twelve 3-inch and two 1-pounder R.F. on this deck.

On the *berth deck* beginning forward are arranged: Lamp room, oil room (starboard), paint room (port); prisons (starboard and port) in forward compartment; band room (starbard), sergeant of marines' room (port); isolation ward (starboard), master-at-arms' room (port); crew's berthing space, in second compartment; chain lockers and crew's berthing space, in third compartment; dispensary (starboard), operating room (starboard); sick-bay bath room (port), and sick bay in fourth compartment; chief petty officers' quarters and wash room (starboard); chief machinists' quarters and wash room (port); chief petty officers' mess room, in fifth compartment; blacksmith shop (forward of No. 1 boiler hatch); firemen's wash rooms (three, forward and aft of No. 2 boiler hatch and forward of No. 3 boiler hatch); evaporator room, between Nos. 3 and 4 boiler hatches; coal bunkers B-101, B-105, B-111, B-115 (starboard); B-102, B-106, B-110, B-114, (port); servants' wash room (aft of No. 4 boiler hatch); engineer's office (forward end of engine-room hatch); ice machine and refrigerating rooms (starboard); machinery work shop (port); paymaster's issuing room (starboard); clothing and small stores, two (starboard); junior officers' pantry (starboard); machinery store room (port); wardroom and junior officers' store rooms (two, port); warrant officers' pantry (port); all within the lower armored casemate, sixth compartment. Junior officers' rooms and mess room (starboard); warrant officers' rooms and mess room (port), in seventh compartment; wardroom rooms, in eighth compartment; captain's store rooms (two, port and starboard); blower room aft, in ninth compartment.

The *protective deck* extends the whole length of the vessel, and its level corresponds with that of the berth deck. This deck is flat amidship and slopes downward to the sides and

ends below the water line. The space between the berth deck and these slopes is utilized for stores, and arranged, beginning forward, as follows: Paint store room, awnings, coaling gear, hospital stores (port and starboard); chief petty officers' stores (port and starboard); coal bunkers B-101, B-102, B-105, B-106, B-111, B-110, B-115, B-114, extend to the protective deck, machinery stores and warrant officers' stores, C-100 (starboard), with a dividing bulkhead at frame 79, provisions, clothing and small stores, C-101 (port) with dividing bulkheads at frames 75 and 79, dry provisions, D-100 and D-101 (port and starboard), ward-room stores, D-104 and D-105 (port and starboard), captain's stores, D-108.

The *upper platforms* are below the protective deck, and beginning at frame 7, these compartments are arranged as follows: General stores; sails and awnings; dry provisions and medical stores (port and starboard); 6-inch magazine; communicating room (central); upper dynamo room (central); 6-inch shell room (port); 6-inch shell room (starboard); ammunition passages (two, port and starboard) through boiler spaces, then one, central, through engine space; 6-inch shell room (port); 6-inch magazine (starboard); 6-inch magazine (athwartship); steering room; steering-gear room.

The *lower platforms* are next below the upper platforms and between frames 13 and 30 forward, arranged as follows: Provisions (athwartship); provisions (port); provisions (starboard); construction stores (starboard); 6-inch magazine (athwartship); 3-inch magazine (port); 3-pounder and 1-pounder magazine (starboard); small arms ammunition room (starboard).

Between frames 83½ and 99, aft, are: Handling room (athwartship); engineers' store rooms (two, port and starboard); 6-inch magazine (starboard); small arms ammunition rooms (two, port and central); 3-pounder magazine (port); ordnance store room (starboard); equipment store room (port); saluting powder room (starboard); navigator's store room (port); ordnance store room (starboard); marines' store room (starboard); equipment store room (port).

The arrangement of the hold is as follows: Trimming tanks (two, first and second compartment); fresh-water tanks (six, third compartment); 3-inch magazine (fourth compartment); electrical store rooms (two, port and starboard); lower dynamo room (central); boiler rooms (four, central); coal bunkers B-5, B-7, B-9, B-11 (starboard) B-6, B-8, B-10, B-12 (port); engine rooms (two, port and starboard); shaft alleys (two, port and starboard); engineers' store rooms (two, port and starboard); 3-inch magazine (central); fresh-water tanks (two, athwartship); trimming tanks (two after compartments).

Hull.—The hull is constructed of steel of domestic manufacture. The transverse frames, 113 in number, stand square to the load line and are spaced 4 feet apart between frames 35 and 84. Forward of 35 and aft of 84, they are spaced 3 feet 6 inches apart. Intermediate frames are worked where necessary for additional strength or stiffness. The vertical keel is 39 inches in depth except forward of frame 7 and abaft frame 102, where it tapers to its connection with the stem and stern post. The plating of this keel is 20 pounds per square foot from frame 23 to frame 84 and 17½ pounds forward of frame 23 and aft of frame 84. Lightening holes are cut in the vertical keel forward and abaft the limits of the double bottoms. There are five longitudinals on each side of the vertical keel and below the protective deck, one stringer each side behind the armor about midway between protective and berth decks. The bilge keels are about 150 feet in length and are secured to the outside plating over the fourth longitudinal. The inner bottom extends from frame 13 to frame 96; the double compartments between frames 56 and 69 are arranged for reserve feed tanks. The stem and sternpost are of cast steel and fitted to receive plating, keel plates and deck connections. The outside bottom plating, below the line of the protective deck, is 23 pounds per square foot for a distance of about 200 feet amidships and is reduced in two reductions to 20 pounds at the ends. The outside plating above

the line of the protective deck is of 30 pounds per square foot behind the armor reducing in five reductions to 20 pounds at the ends.

COAL BUNKER CAPACITIES.

Measured to 6 inches below bottom of deck beams, at 43 cubic feet per ton.

Bunkers above protective deck.	Cubic feet.	Tons.	
B-101	4,347.10	101.09	
B-102	4,347.10	101.09	
B-105	4,239.61	98.59	
B-106	4,239.61	98.59	
B-110	4,370.22	101.63	
B-111	4,370.22	101.63	
B-114	4,272.31	99.35	
B-115	4,272.31	99.35	
	Total,		801.32 tons.
Bunkers below protective deck.			
B-5	3,276.65	76.20	
B-6	3,276.65	76.20	
B-7	4,687.97	109.02	
B-8	4,687.97	109.02	
B-9	5,080.95	118.16	
B-10	5,080.95	118.16	
B-11	5,177.94	120.41	
B-12	5,177.94	120.41	
	70,905.50		847.58
	Total		1,648.90

DOUBLE BOTTOM AND RESERVE FEED WATER TANK CAPACITIES.

Compartment.	Gallons.	Salt Water, Tons.	Fresh Water, Tons.
A-93	7,013.10	26.79	26.04
A-94	7,013.10	26.79	26.04
A-95	5,574.77	21.29	20.70

A-96	5,574.77	21.29	20.70
A-97	20,343.80	77.71	75.55
A-98	15,571.79	59.48	57.83
A-99	12,839.35	49.04	47.79
B-73	6,881.67	26.28	25.55
B-74	6,881.67	26.28	25.55
B-75	5,814.58	22.21	21.59
B-76	5,814.58	22.21	21.59
B-77	6,047.95	23.10	22.46
B-78	6,047.95	23.10	22.46
B-79	6,187.98	23.63	22.98
B-80	6,187.98	23.63	22.98
B-81	6,252.16	23.88	23.22
B-82	6,252.16	23.88	23.22
B-83	6,236.97	23.82	23.16
B-84	6,236.97	23.82	23.16
B-85	6,135.47	23.43	22.78
B-86	6,135.47	23.43	22.78
B-87	7,538.05	28.79	27.99
B-88	7,538.05	28.79	27.99
B-89	16,102.94	61.51	59.80
B-90	13,664.16	52.19	50.74
B-91	14,160.09	54.09	52.59
B-92	14,533.49	55.51	53.97
B-93	14,726.03	56.22	54.69
C-91	7,202.57	27.51	26.75
C-92	7,202.57	27.51	26.75
C-93	6,592.87	25.18	24.48
C-94	6,592.87	25.18	24.48
C-95	5,787.73	22.11	21.49
C-96	5,787.73	22.11	21.49
C-97	15,388.23	58.78	57.14
C-98	13,317.02	50.87	49.54
C-99	12,529.37	47.86	46.53
D-98	8,004.80	30.57	29.75
D-99	5,251.86	20.06	19.50

B-94	7,349.10	28.07	27.29	(Reserve feed)
B-95	7,349.10	28.07	27.29	(Reserve feed)
B-96	7,220.67	27.58	26.81	(Reserve feed)
B-97	7,220.67	27.58	26.81	(Reserve feed)
B-98	8,785.19	33.55	32.62	(Reserve feed)
B-99	8,785.19	33.55	32.62	(Reserve feed)
Total,	389,674.59	1,488.33	1,447.24	

FRESH WATER TANK CAPACITIES.

Compartment.	Tank.		Gallons.	Tons.
A-3	No. 1 between 13 -16,	starboard	915	3.39
A-3	No. 2 between 13 -16,	port	915	3.39
A-3	No. 3 between 17 -19½,	starboard	1,127	4.16
A-3	No. 4 between 17 -19½,	port	1,127	4.16
A-3	No. 5 between 19½-22½,	starboard	1,593	5.89
A-3	No. 6 between 19½-22½,	port	1,593	5.89
D-8	No. 7 between 96 -99,	starboard	1,488	5.50
D-8	No. 8 between 96 -99,	port	1,488	5.50
Gravity, upper deck, forward			1,200	4.44
Gravity, upper deck, aft			400	1.48
Gravity, upper deck, aft			400	1.48
	Total,		12,246	45.28

TRIMMING TANK CAPACITIES.

Compartment.	Gallons.	Salt Water, Tons.	Fresh Water, Tons.
A-1	13,480.98	51.49	50.06
A-2	17,169.89	65.58	63.76
D-9	11,057.98	42.24	41.06
D-10	12,012.13	45.88	44.61
Total,	53,720.98	205.19	199.49

Coffer Dams.—A coffer-dam belt about 30 inches wide extends the length of the ship at sides, from frame 7 forward

to frame 109 aft, occupying the space on the slope of the protective deck outboard and rising to the level of the berth deck. The compartments in this belt are: A-101, A-104, A-105, A-108, A-109, A-112, A-113, B-103, B-104, B-107, B-108, B-112, B-113, B-116, B-117, C-102, C-103, D-102, D-103, D-107, D-109, D-110. These coffer dams are packed with fireproofed, cornpith cellulose.

Armor and Protection.—The side armor extends between frames 35 and 84, abreast of the boiler and engine compartments, and is 4 inches thick by 7 feet 6 inches in height, of which 3 feet 5 inches is above the load water line. The upper edge reaches the berth-deck level.

Nickel-steel plates 1½ inches thick on the flats and 2½ inches thick on the slopes are worked on the 20-pound plating of the protective deck, making a total thickness of 2 inches on the flats and 3 inches on the slopes.

The *upper casemate* extends from gun-deck to main-deck level, and has 4-inch side armor and 3-inch athwartship armor. The upper casemate extends from frame 41 to frame 73, a distance of 128 feet. The forward athwartship armor coincides with the after end of No. 1 boiler-hatch enclosure, while the after athwartship armor coincides with the forward end of engine-room hatch. The eight 6-inch guns on the gun deck are within this casemate.

The *lower casemate* extends from berth-deck to gun-deck level, the side armor being 4 inches thick, and the athwartship armor at ends being 3 inches thick. The lower casemate extends from frame 35 to frame 84, a distance of 196 feet.

The four 6-inch guns on the main deck are protected by sponsons of 4-inch armor, extending from the top of the upper casemate to the upper-deck level.

The twelve 3-inch guns on the gun deck outside of the upper casemate are protected by 2-inch nickel-steel port plates.

An armored conning tower, elliptical in shape with its long axis athwartships, is located at frame 28. The entrance is at the afterside and is protected by a concentric shield plate. Both the tower and shield are 5 inches thick.

MAIN BATTERY.

Fourteen 6-inch, 50-caliber, R.F. guns.

Number.	Location.	Train for'd and abaft beam.		
			For'd.	*Aft.*
Nos. 1 and 2,	Gun deck, frame 44,	S. & P.,	85°	35°
Nos. 3 and 4,	Gun deck, frame 52½,	S. & P.,	77°	35°
Nos. 5 and 6,	Gun deck, frame 61½,	S. & P.,	35°	77°
Nos. 7 and 8,	Gun deck, frame 70,	S. & P.,	35°	85°
No. 9,	Main deck, frame 23,	central,	90°	55°
Nos. 10 and 11,	Main deck, frame 44,	S. & P.,	90°	33°
Nos. 12 and 13,	Main deck, frame 70,	S. & P.,	33°	90°
No. 14,	Main deck, frame 91,	central,	55°	90°

SECONDARY BATTERY.

Eighteen 3-inch, 50-caliber, R.F. guns.

Nos. 1 and 2,	Gun deck, frame 13½,	S. & P.,	90°	30°
Nos. 3 and 4,	Gun deck, frame 23½,	S. & P.,	77°	43°
Nos. 5 and 6,	Gun deck, frame 31½,	S. & P.,	76°	44°
Nos. 7 and 8,	Gun deck, frame 82½,	S. & P.,	44°	76°
Nos. 9 and 10,	Gun deck, frame 91½,	S. & P.,	43°	77°
Nos. 11 and 12,	Gun deck, frame 101½,	S. & P.,	30°	90°
Nos. 13 and 14,	Main deck, frame 52½,	S. & P.,	60°	60°
Nos. 15 and 16,	Main deck, frame 57,	S. & P.,	60°	60°
Nos. 17 and 18,	Main deck, frame 61½,	S. & P.,	60°	60°

Twelve 3-pounder, semi-automatic guns.

Nos. 1 and 2,	Upper deck, frame 28½,	S. & P.	
Nos. 3 and 4,	Upper deck, frame 31,	S. & P.	
Nos. 5 and 6,	Upper deck, frame 45,	S. & P.	
Nos. 7 and 8,	Upper deck, frame 69,	S. & P.	
Nos. 9 and 10,	Upper deck, frame 81½,	S. & P.	
Nos. 11 and 12,	Upper deck, frame 85½,	S. & P.	

Eight 1-pounder, heavy, R.F. guns.

Nos. 1 and 2,	Gun deck, frame 110½,	S. & P.	
Nos. 3 and 4,	Upper deck, frame 36,	S. & P.	
Nos. 5 and 6,	Upper deck, frame 78½,	S. & P.	
Nos. 7 and 8,	After bridge, frame 82,	S. & P.	

Six .30-caliber guns.

Nos. 1 and 2, Forward bridge, S. & P., (Colt automatic)
Nos. 3 and 4, Forward bridge, S. & P., (Gatling)
Nos. 5 and 6, After bridge, S. & P., (Colt and automatic)

Ammunition Hoists.—The twenty-seven electrically-operated ammunition hoists are driven by reversible motors operating at about 1,000 R.P.M. Frictional slip gears are arranged between the motors and hoists to prevent damage being done to them in case of sudden stoppage of the hoists. Stops are provided to prevent ammunition falling back in the hoists. Hand operating gear is also fitted.

Anchors.—The following anchors are supplied:

Two National patent anchors, weight (two) 12,000 pounds.
One sheet, Navy Standard, weight, 13,000 pounds.
One stern, Navy Standard, weight, 3,500 pounds.
One stream, Navy Standard, weight, 2,500 pounds.
One kedge, Navy Standard, weight, 800 pounds.
One kedge, Navy Standard, weight, 400 pounds.
Eighteen small boat anchors, varying from 50 to 120 pounds.

MAIN ENGINES.

There are two four-cylinder, triple-expansion engines of the vertical, inverted, direct-acting type, designed to indicate 21,000 horsepower at about 133 revolutions per minute with a steam pressure of 250 pounds at the high-pressure cylinder. The engines are placed abreast, in separate watertight compartments, with a watertight communicating door in the center-line bulkhead between the two engine rooms. The arrangement of the cylinders, beginning forward, is forward low-pressure, high-pressure, intermediate-pressure and after low-pressure. The forward low-pressure and high-pressure cylinders are securely bolted together, also the intermediate and after low-pressure cylinders, while facings are left on the after side of high-pressure cylinder fitting removable brackets on intermediate cylinder casings, to prevent side motion and at the same time to allow for expansion in a fore-and-aft di-

rection. All cylinders are of cast iron, and are steam jacketed around the working liners. The heads of the intermediate and low-pressure cylinders are also jacketed. The cylinders for each engine are supported by twelve forged-steel columns, 5¾ inches in diameter, flanged at top and bottom, for bolting to cylinders and bed plates. For each athwartship pair of columns there are two diagonal braces, rectangular in section, forming an X-brace. The outer ends are tongued and fitted to grooves at top and foot of columns and bolted. The two braces are secured together at their centers by two keys, also by a horizontal athwartship tie rod, 2½ inches in diameter, passing through columns and braces, with sleeve distance pieces. The framing is further stiffened by fore-and-aft horizontal and diagonal tie rods secured to columns. There are two cast-steel strong backs bolted to outboard row of columns to which the crosshead guides are bolted. The engine *bed-plates* are made of cast steel, each in three sections, with flanges for bolting together. Each section consists of two longitudinal girders and two cross girders, all of I section, and well stiffened by ribs, and finished for crank-shaft bearings, columns and turning engine. The bed plates are secured to the engine keelsons by body-bound forged-steel bolts, with forged-steel washers fitted at each holding-down bolt between bed plate and engine keelson. Yellow pine strips are fitted between washers. The *valve chest* of each high-pressure cylinder is fitted with one, and the intermediate and low-pressure cylinders with two single-ported piston valves. The valves are cast iron, made hollow for the passage of steam to the ends of the valve chest, and are all provided with balance pistons, the cylinders of which form part of the upper covers of the valve chests. Each end of the piston valves is fitted with a single cast-iron packing ring held in place with a forged-steel follower, and made to permit of contraction but not expansion. The *piston valves* for all cylinders are operated by the Stephenson double-bar link motion with adjust-

able cut-off. The H.P. *steam pistons* are made of cast iron, the intermediate and low-pressure pistons of cast steel, conical in shape, and finished all over. Wrought-steel followers hold in place hard cast-iron packing rings, one wide ring being fitted in each of the high and intermediate-pressure pistons, and two narrow rings in the low-pressure pistons. The *piston rods* are made of forged nickel-steel, oil tempered and annealed, with axial holes the entire length of the rods. The rods are accurately ground and polished to an outside diameter of 8 inches. The *crossheads* are made of forged steel, "Class A No. 1." The pins extend on each side of the crosshead and are forged in one piece with it. The *slippers* are cast steel, bolted to the outboard side of the crosshead, and the sliding surfaces are faced with Navy standard white metal. Cast-iron *crosshead guides* made hollow for the circulation of cooling water, are bolted to the cast-steel strong backs on the back engine columns, and at their tops to the cylinder casings. The backing guides are also made of cast iron, and bolted to flanges on the go-ahead guides. The *connecting rods* are of forged nickel-steel, oil tempered and annealed, with axial holes the entire length. The upper ends are forked to span the crossheads and carry the crosshead brasses. Both the crosshead and crank-pin brasses are lined with Navy standard white metal. The *eccentrics* are made in two parts, the larger part being made of cast iron and the smaller of forged steel. The two parts are fitted together and secured by 2-inch steel bolts. Each backing eccentric is securely keyed to the crank shaft, and each go-ahead eccentric is secured to the shaft by a key, with adjusting pieces between the key and eccentric. The *eccentric straps* are of composition lined with Navy standard white metal. The *eccentric rods* are nickel-steel, with forked upper ends to span the links, and T-heads at lower end for bolting to eccentric straps. The *main links* are nickel-steel, double bar, with pins forged on for drag links and eccentric rods.

The go-ahead half of each link is divided into ten equal divisions, and graduated, 0 being the mid-position of the link and 10 the position when in full gear. There is a 14-inch *balanced throttle* valve fitted on each engine, balanced by a piston working in a cylinder forming a part of the throttle-valve casing, steam being admitted to the back of the balance piston. To further insure ease in closing this size of valve against such high pressure a toggle gear is fitted. On the stem at handwheel is an index showing the fractional valve opening in tenth parts of full opening.

All the *crank, line, thrust, propeller* and *stern-tube shafts* are forged nickel-steel with axial holes their entire length. The crank shafts are made in two sections, with two cranks on each section, placed opposite each other. The forward low-pressure and high-pressure cranks are opposite, also the intermediate and after low-pressure cranks, the second pair being at right angles with the first, the sequence of the cranks being, high-pressure, intermediate-pressure, forward low-pressure and after low-pressure. Each complete crank shaft has six main bearings, each thrust shaft two steady bearings, and each line shaft one spring bearing. Each stern-tube shaft is supported by bearings at each end of stern tube, and the propeller shaft has one strut bearing. The stern-tube and strut bearings are lined with lignum vitae, end grain exposed, while the spring, steady and main bearings are lined with white metal. The *thrust bearing* is of the horse-shoe type. The body is of cast iron, and the horse shoes of cast steel. The end and side walls of the body form an oil reservoir, in which the lower part of the collars turn. The reservoir is fitted with a circulating coil for cooling water. The horse shoes are made hollow for the circulating or cooling water and faced with white metal. The *outboard coupling* between stern-tube and propeller shafts consists of a forged-steel sleeve, bored taper for ends of shafts and secured by longitudinal and cross keys. The coupling between the stern-tube and line shafts consists of a collar securely keyed

to forward end of stern-tube shaft by four keys. Aft of this collar another collar of the same diameter, made in halves, is secured to the forward collar and to the after coupling flange of the line shafts by eight $3\frac{9}{16}$-inch fitted bolts. The *reversing gear* for each main engine consists of a steam cylinder and oil-controlling cylinder, acting on arms fixed on the reversing shaft by means of connecting links. The reversing engine is bolted to the high-pressure cylinder casing on the inboard side of the main engines. The valve of the steam cylinder is of the piston type, of composition, and works in a composition-lined valve chest. The by-pass valve on the oil cylinder is worked by a continuation of the stem of the steam piston valve. These valves are worked by a floating lever, the primary motion being derived from a hand lever at the working platform and the secondary motion from the reversing shaft. There is a stop cock in the by-pass pipe of the oil cylinder, and a pump for reversing by hand connected to the oil cylinder. The *reverse-shaft bearings* are bolted near the top of the inboard main-engine columns. The shaft is of high-grade steel, and arms of cast steel. Diameter of steam cylinder, 15 inches; oil cylinder, $7\frac{1}{2}$ inches, with a common stroke of $21\frac{1}{2}$ inches.

A vertical double-cylinder *turning engine* is located on the outboard side of each main engine and bolted to the main-engine bed plate. A worm on the turning-engine crank shaft drives a vertical shaft, a worm on the lower end of which meshes with a worm wheel secured to the periphery of the coupling flanges between the two sections of main-engine crank shaft. The cylinders are 6 inches in diameter and 6 inches stroke. A ratchet is fitted for hand turning.

Cylinders, number for each engine	4
H.P., diameter, inches	36
I.P., diameter, inches	$59\frac{1}{2}$
L.P. (two), diameter, inches	69
stroke, inches	45
Valves, H.P. (one piston valve for each cylinder):	
diameter, top, inches	$22\frac{1}{16}$
bottom, inches	22

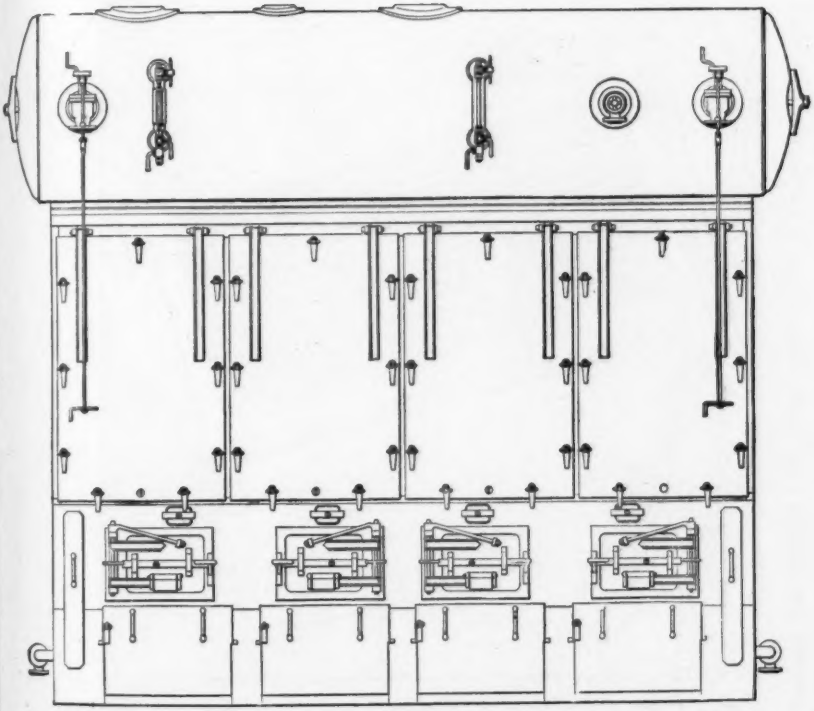
Valves, I.P. (two piston valves for each cylinder):	
diameter, top, inches	25 9/16
bottom, inches	25 1/4
L.P. (two piston valves for each cylinder):	
diameter, top, inches	25 9/16
bottom, inches	25 1/4
Valve stems, diameter through stuffing boxes, inches	3 1/2
valves, inches	2
at balance pistons, inches	1 7/8
Balance-piston cylinders, diameter, inches, H.P.	7
I.P.	6 1/2
L.P.	10 3/8
Main steam pipe, diameter at throttle, inches	14
Exhaust pipe to condenser, diameter, inches	27
Cylinder walls, H.P., I.P. and L.P., thickness, inches	2 1/8, 1 5/8, 1 1/4
liners, H.P., I.P. and L.P., thickness, inches	1 1/8, 1 1/2, 1 3/8
Jacket space, inch	3/4
Valve-chest liners, thicknes, inches	1 3/8
Cylinder relief valves (one each end each cylinder), diameter, inches.	4
Piston rods, diameter, inches	8
axial hole, H.P., diameter, inches	1 1/2
I.P., diameter, inches	4
Piston rod, axial hole, F.L.P., diameter, inches	5
A.L.P., diameter, inches	4
Connecting rods, section at upper end, diameter, inches	8 1/2
lower end, diameter, inches	9 3/8
axial hole, H.P., diameter, inches	1 1/2
I.P., diameter, inches	3
L.P., diameter, inches	5
center to center, inches	90
Crank-pin bolts (two), diameter, inches	4 1/2
Crosshead pins, diameter, inches	10 1/2
length (each side of crosshead)	10 1/2
axial hole, H.P., diameter, inches	1 1/2
I.P. and L.P. taper, diameter, inches	6 to 2
Crossheads, wearing surface, ahead, inches	24×30 1/2
backing (two), inches	9×30 1/2
Crank shaft, sections	2
diameter, inches	18
axial hole, diameter, inches	11 1/2
coupling flange, diameter, inches	32
thickness, inches	3 1/4
coupling bolts (eight), diameter, inches	3 9/16
journals, diameter, inches	18
(two) length, inches	32 1/2
(two) length, inches	22 1/2
(one) length, inches	31
(one) length, inches	27 1/2
length over all, feet and inches	35–03

Crank pins, diameter, inches	18¼
length, inches	22
diameter of axial hole, inches	12
webs, width, inches	20
thickness, inches	11¾
Thrust shaft, diameter, inches	17¼
axial holes, diameter, inches	11½
collars, number, each shaft	13
diameter, inches	25½
thickness, inches	2
space between, inches	4
length, feet and inches	12-06
Line shaft, diameter, inches	17
axial hole, diameter, inches	11½
length, feet	22
coupling flange, diameter, inches	32
thickness, inches	3¼
bolts (eight), diameter, inches	3 9/16
Stern-tube shaft, diameter, inches	18
axial hole, diameter, inches	11
length, feet and inches	32-08⅜
Propeller shaft, diameter, inches	18
axial hole, diameter, inches	11
length, feet and inches	36-05

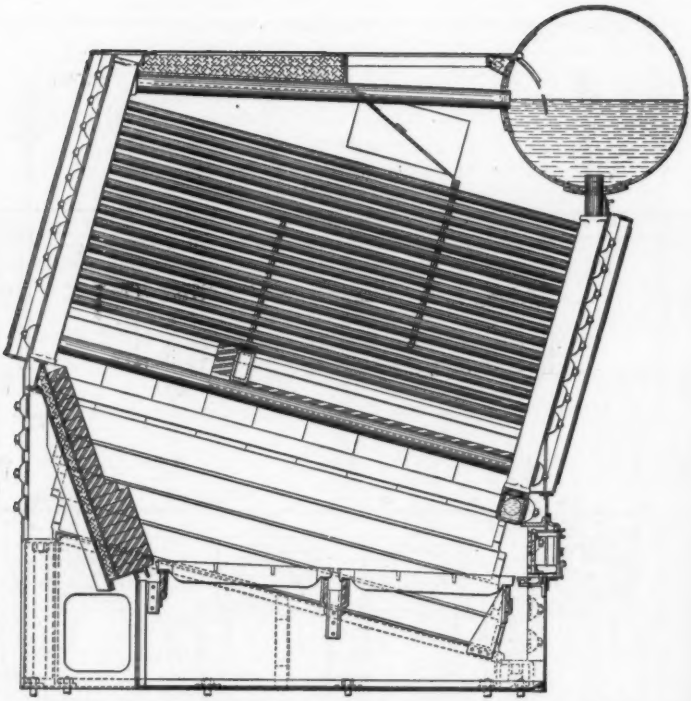
CYLINDER CLEARANCES, CRUISER NO. 20, THE *ST. LOUIS.*

Engine.	Cylinder.	Volume of cylinder, cubic inches.	Clearance, in cubic inches.	Per cent. of cylinder volume.	Linear clearance, inches.	Ride of ring, over counterbore, inches.
Starboard....	H.P ...Top.....	45,805	13,860	30.2	9/16	5/16
	Bottom.....	43,543	12,705	29.1	1/4	1/16
	I.P.....Top.....	125,123	27,951	22.3	9/16	5/16
	Bottom	122,861	29,730	24.2	1/4	1/16
	F.L.P.Top.....	168,268	30,492	18.1	9/16	5/16
	Bottom	166,006	35,066	21.1	1/4	1/16
	A.L.P..Top.....	168,268	30,492	18.1	7/16	5/16
	Bottom	166,006	35,066	21.1	1/4	1/16
Port.....	H.P....Top	45,805	13,860	30.2	9/16	1/4
	Bottom.....	43,543	12,705	29.1	1/4	1/16
	I.P.....Top	125,123	27,951	22.3	9/16	1/4
	Bottom.....	122,861	29,730	24.2	1/4	1/16
	F.L.P..Top	168,268	30,492	18.1	9/16	5/16
	Bottom.....	166,006	35,066	21.1	7/16	5/16
	A.L.P..Top.....	168,268	30,492	18.1	7/16	5/16
	Bottom.....	166,006	35,066	21.1	7/16	5/16

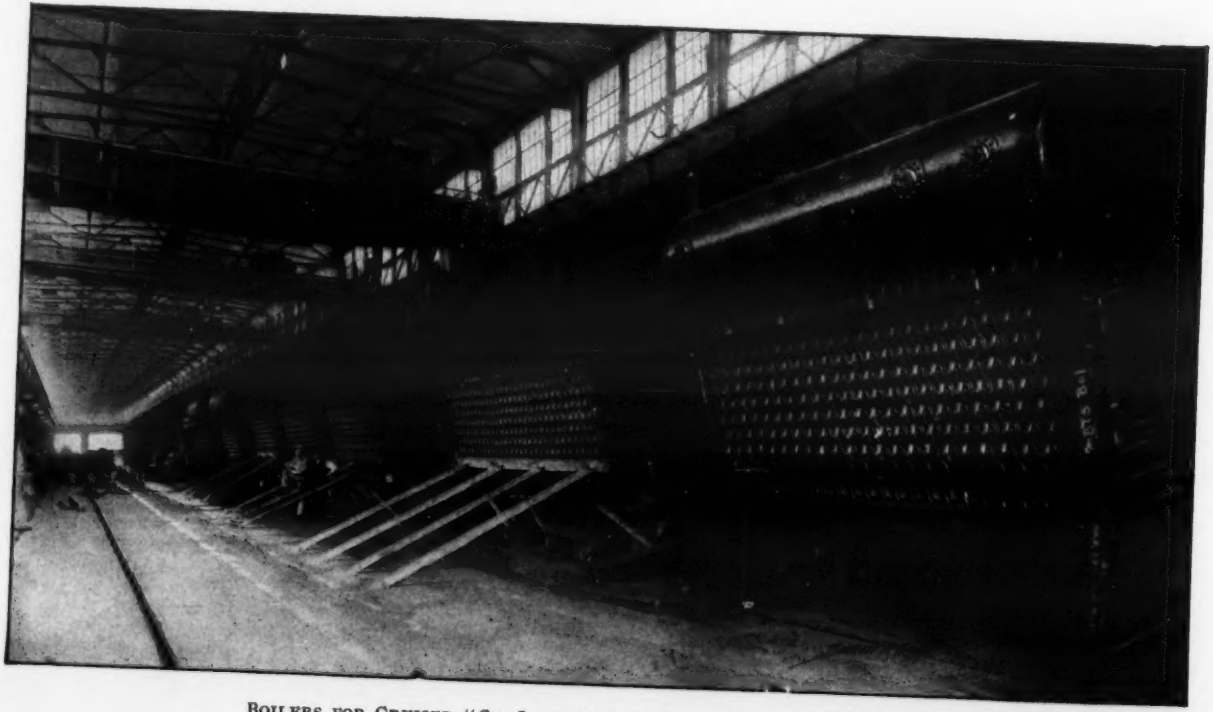
10 per cent. was added to actual measurement of volumetric clearance for bottoms of I.P. and L.P. cylinders for the reason that the air could not be liberated.



FRONT VIEW OF BABCOCK & WILCOX BOILER AS FITTED INTO THE CRUISER "ST. LOUIS."



SIDE VIEW OF BABCOCK & WILCOX BOILER AS FITTED INTO CRUISER "ST. LOUIS."



BOILERS FOR CRUISER "ST. LOUIS" ERECTED IN SHOP BEFORE SHIPMENT.

FIREROOM OF CRUISER "ST. LOUIS."

BOILERS.

There are sixteen Babcock and Wilcox water-tube boilers, placed in four watertight compartments, and designed for a working pressure of 265 pounds per square inch. The two forward boilers in each compartment face aft, while the two after boilers face forward, making roomy fire-rooms. The description of the Babcock & Wilcox boilers by Lieutenant R. K. Crank, U. S. Navy, Vol. XVI, November, 1904, of the JOURNAL, applies to the boilers of the *St. Louis,* with the exception that the boilers of the *St. Louis* have no middle water leg dividing the furnaces.

Number of boilers	16
furnaces	16
Steam drum, inside diameter, inches	42
thickness, inch	00$\frac{1}{32}$
Grate, length, feet and inches	7–00
width, feet and inches	12–06
surface, one boiler, square feet	87.5
all boilers, square feet	1,400
Heating surface, one boiler, square feet	4,000
all boilers, square feet	64,000
Ratio of grate surface to heating surface	45.71
Elements, one boiler (21 intermediates, 2 sides)	23
Tubes, one boiler, 2-inch, No. 8 B.W.G., 9 feet 2 inches long	722
4-inch, No. 6 B.W.G., 9 feet 2 inches long	23
7 feet 3 inches long	23
Safety valves (one triple type each boiler), total	48
diameter of each valve, inches	3
set to blow at pounds	283.25
Sentinel valves (one each boiler), diameter, inch	00$\frac{3}{4}$
Smoke pipes (four) diameter, inside, feet and inches	8–00
area, each, square feet	50.265
total, square feet	201.06
height above grates, feet and inches	92–06
Width of air space in grates, inch	00$\frac{1}{2}$
metal in grates, inch	00$\frac{1}{2}$
percentage of air space	48
Floor space, length one boiler, feet and inches	10–01
width one boiler, feet and inches	13–08$\frac{1}{2}$
height one boiler, feet and inches	13–04$\frac{1}{2}$
Weight of water, one boiler, pounds	13,384
one boiler complete, empty, pounds	80,450
with water and fittings, pounds	93,834

Boiler Stop Valves.—The boiler stop valves are steam opening and closing valves, and are the invention of Commander A. B. Willits, U. S. N. In this design the valve and actuating piston are in the same chamber, and only one stuffing box is necessary. The operating cock shown in cut one for each boiler, for steam opening and closing, is located in the next compartment aft the one containing said set.

The operating cocks for boilers in No. 1 boiler room are located in No. 2 boiler room; for No. 2 in No. 3; for No. 3 in No. 4; for No. 4 in engine rooms.

These valves work highly satisfactory in every respect. A full description of this valve is given in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XVI, No. 4, by the inventor.

STEAM PIPING.

The main steam pipes are made up of seamless-drawn steel pipes with forged-steel flanges. The pipes are rolled into the flanges, beaded over to fit the recesses and riveted to the flanges. The main steam piping is arranged symmetrically in two systems, one for each side of the ship, beginning in No. 1 fireroom at 7 inches in diameter, increasing to 10, 12 and 14 inches in diameter, successively as they lead aft through Nos. 2, 3 and 4 firerooms. In No. 4 fireroom there is a 9-inch cross connection connecting the two systems. Slip joints are placed at each bulkhead to provide for expansion. Abaft the engine-room bulkhead the lead to the throttle is 14 inches in diameter. All stop valves in the main steam line are fitted with by-pass valves to relieve the valves when jammed on their seats. The auxiliary steam-pipe system in the firerooms parallels the main system and passes on forward to the dynamo rooms, with a cross connection in the forward boiler compartment. In the engine rooms it leads around the outboard and after sides of the engine rooms and through the middle-line bulkhead. A 6-inch pipe with a stop valve

connects the main steam with the auxiliary steam pipe in each engine room. The auxiliary steam pipe has a diameter of 6 inches through the engine rooms, 7 inches through the firerooms and 6½ inches in the dynamo rooms. Each line connects with the main stop valves on all boilers on its own side of the ship. Steam for auxiliary machinery forward of the boiler compartments and to steam whistle is taken from the 6½-inch cross connection in No. 1 fireroom. Steam to fireroom pumps, boiler-tube cleaning, sea chests, blower and ash-hoisting engines is taken from the auxiliary steam lines in the firerooms.

Exhaust Piping.—Throughout the engine and firerooms the auxiliary exhaust piping parallels the auxiliary steam piping, with cross connections in engine rooms and No. 1 fireroom. The exhaust steam leads to main and auxiliary condensers, low-pressure receivers, feed-water heaters, and into the atmosphere through the after escape pipe. At each connection with a condenser there is a stop valve and spring relief valve. The object of the relief valve is to produce a back pressure when the exhaust steam is used in the low-pressure receivers or feed-water heaters. Two stop valves are fitted at each connection with the escape pipe to minimize the chance of air leak. Exhaust steam from the dynamo engine is led into the auxiliary condenser in the dynamo room, and to the auxiliary exhaust pipe. Separators are fitted in dynamo engine exhausts.

DRAINS AND TRAPS.

All places where condensed steam can accumulate are drained. Main-engine cylinder and valve-chest drains lead to the main condensers with branches to bilge. Dynamo cylinder and valve-chest drains to auxiliary condenser in the lower dynamo room with branches to bilge. Steam separators, cylinder jackets, steam pipes and evaporator coils drain through traps to feed tanks. Drains from heating system and galley cookers lead to traps which discharge into feed tank, auxiliary condenser, or to bilge at

will. Drain from pantry steam leads to the auxiliary exhaust line. Drains from auxiliary cylinders are led to the auxiliary exhaust and to bilge. The Kinney, open-float, automatic steam trap is used. In this trap the by-pass is contained in the body of the casting, so no extra valves or piping are necessary to by-pass the condensed steam when overhauling the traps.

SCREW PROPELLERS.

The propellers have each three blades, adjustable pitch, and made of manganese-bronze. The blades are bolted to the hubs by rolled manganese-bronze tap bolts, and secured by lock plates. The hubs are bored to fit the taper on the end of the shaft, are fitted with two keys, and held by a nut screwed on the end of the shaft against the hub and locked in place. The blades, hubs and caps are finished smooth and burnished. The propellers turn inboard when going ahead.

Number of blades	3
Diameter of propeller, feet	18
hub, feet and inches	4–07½
Length of hub, inches	45
Pitch of starboard propeller, feet and inches	19–05
port propeller, feet and inches	19–04.22
Helicoidal area, each, square feet	91.5
Disc area, each, square feet	231.9
Projected area, each, square feet	76.74
Center of hub above lowest point of keel, feet	9
Top of propeller immersed (22 feet 6 inches draught), inches	54

MAIN CONDENSERS.

There is one main condenser in each engine room, cylindrical in shape, with an inside diameter of shell of 7 feet. The shells are of wrought steel, Class B boiler plate, ⅜-inch thick, with double butt joints and circumferential angle-bar stiffeners, with angle bars riveted around each end to form flanges for securing the water chests. The water chests are of composition, tube sheets of rolled Muntz metal, and tubes seamless-drawn and of a mixture

70 copper, 29 zinc, 1 tin, and ⅝-inch outside diameter. The forward chest, being the one for the entrance and exit of circulating water, has a division plate fitted with valves which, when open, allow the circulating water to pass overboard direct when pumping from the bilge. These valves are worked by a lever on the outside of the condenser. The tubes are packed with cotton-tape packing. The glands are counterbored, leaving a shoulder at outer ends to prevent the tubes from crawling, and are threaded with 18 threads per inch. The tubes are supported at two intermediate points by ferrules inserted in supporting plates. Baffle plates are fitted to direct the steam over all the tubes. There are 5,714 tubes in each condenser, No. 18 B. W. G. in thickness, 14 feet and 4 inches long between tube sheets, spaced $\frac{15}{16}$-inch between centers. The cooling surfaces for each condenser is 13,400 square feet, measured on outside of tubes. The water chests are fitted with rolled-zinc plates. The ratio of total cooling surface to total heating surface is 1 to 2.38. These condensers are covered with hair felt and lagged with galvanized iron.

MAIN AIR PUMPS.

For each main engine there is one double, vertical, single-acting Davidson air pump, with two inverted steam cylinders 14 inches in diameter and a common stroke of 18 inches. The air-pump buckets are 33½ inches in diameter, and each has eighteen 5-inch valves composed of three flat discs of rolled manganese-bronze plates $\frac{1}{32}$-inch thick. The suction and discharge-valve decks each contain 20 similar valves. The suction and discharge openings are 12½ and 10 inches, respectively.

MAIN CIRCULATING PUMPS.

For each main condenser there is one double-inlet, centrifugal, circulating pump driven by a vertical compound engine with cranks at right angles. The H.P. cylinder has a diameter of 10 inches, the L.P. cylinder 18 inches, with

a stroke of 9 inches. The pump runner is 48 inches in diameter and width of runner at periphery, 5¾ inches The suction and discharge nozzles are each 20½ inches in diameter. Each pump is fitted to draw from the sea, the main drain, and the engine-room bilge, and each of the three valves has a screw-down valve placed so as to open toward the pump. The three valves are interlocked in such a way that only one can be opened at a time. On the inboard side of the sea valve, between the sea valve and the pump, is fitted a safety sluice gate. Each of these pumps is capable of discharging 12,000 gallons per minute from the bilge.

FEED AND FILTER TANKS.

There is in each engine room a feed and filter tank of 5,000 gallons capacity. The forward end of each tank is arranged as a filter chamber, *loofa* being the filtering material used. A series of vertical partial partitions divide the filter chamber into smaller chambers. The entering water flows into the first chamber over the top of a plate and leaves the last chamber in like manner, insuring that the filter chambers are always filled and the filtering material submerged. Each compartment of the feed and filter tank is provided with a vapor pipe, connecting with the after escape pipe. Each tank is fitted with an overflow pipe leading to the bilge and so arranged that any water passing through it may be seen. The tanks are covered with hair felt and lagged with galvanized iron.

FEED PUMPS.

There are four main and four auxiliary feed pumps, of the Davidson vertical, double-acting, plunger type, all the same size, 14 inches by 10 inches by 15 inches. The main feed pumps are located in the engine rooms, and draw from the connecting pipe between main feed tanks, and deliver into the main feed pipes, either through the feed-water heaters, or by by-passing them. One auxiliary feed

pump is located in each fireroom and arranged to draw from the connecting pipe between feed tanks in the engine rooms, from the auxiliary drain, from the sea, from the fireroom bilge of its own compartment and from the bottom blow pipes of all boilers in its own compartment. Each delivers into a fore-and-aft feed main leading to all boilers on its own side of the ship, into the fire main, and overboard through a sea valve in its own compartment.

FEED-WATER HEATERS.

There is one feed-water heater in each engine room, located on the discharge side of the main feed pumps. The shells are made of copper $\frac{5}{16}$-inch thick and 35 inches in diameter. In each heater there are 976 $\frac{5}{8}$-inch Muntz-metal tubes, 6 feet long between tube sheets, No. 14 B. W. G. in thickness, expanded into each tube sheet, making a heating surface of 958 square feet. The tube sheet at one end is of Muntz metal and is bolted to a composition flange on the end of the shell, and at the other end the tube sheet is a composition casting forming part of an expansion joint. The feed water, on its way to the boilers, enters a nozzle at the bottom of the movable head, and passes out at the top of the fixed head. Steam, from the auxiliary exhaust line, enters near the top of the shell, and after passing around the tubes is drained to the main and auxiliary condensers and to feed tanks. The heaters are covered with magnesia and lagged with galvanized iron.

HOT-WELL PUMPS.

A Davidson vertical, single-cylinder hot-well pump, 12 inches by 14 inches by 12 inches is located in each engine room, connected to draw from main feed tanks, main air-pump suction pipes, reserve-feed tanks, and from the sides of the ship above the water line for taking in fresh water. There is a one-inch connection between the hot-well pump suction from reserve-feed tanks and the main air-pump suction. This will avoid the use of the hot-well pump to

draw from the reserve feed tanks for "making-up" feed. The hot-well pumps discharge into the feed tanks, reserve-feed tanks, and to connecting pipe between main feed tanks.

FIRE AND BILGE PUMPS.

There is a Davidson fire and bilge pump in each engine room, of the same type as the hot-well pumps. These pumps draw from the sea, the auxiliary drain, the drainage manifold, the engine-room bilge, the crank pits, and also connects to the suction of the shaft bilge pump. Delivers to the fire main and overboard.

SHAFT BILGE PUMPS.

A crank pin on the forward end of each main-engine crank shaft drives a pump which draws from the bilge and crank pits and discharges overboard.

AUXILIARY CONDENSERS.

There is an auxiliary condenser with combined horizontal air and circulating pump in each engine room, connected through the auxiliary exhaust pipe to all the auxiliary machinery. The shells are wrought-steel plate, heads composition, and tubes plates of Muntz metal. Diameter of air cylinder, 8 inches; circulating-water cylinder, 12 inches; steam cylinder, 7 inches, with a common stroke of 12 inches.

Diameter of shell, inside, inches	28
Thickness of shell, inch	00$\frac{1}{2}$
Tubes, diameter, inch	00$\frac{5}{8}$
length between tube sheets, feet and inches	6–03$\frac{1}{2}$
number in one condenser	601
cooling surface, each condenser, square feet	620
total square feet	1,240

There is one auxiliary condenser in the lower dynamo room with combined horizontal air and circulating pump, for the exclusive use of the dynamo engines. Diameter of air cylinder, 12 inches; circulating-water cylinder, 16 inches; steam cylinder, 10 inches, with a common stroke of 12 inches.

Diameter of shell, inside, inches	36
Thickness of shell, inch	00½
Tubes, diameter, inch	00⅝
length between tube sheets, feet and inches	7–04
number	1,012
cooling surface, square feet	1,214

FORCED DRAFT.

The closed-fireroom system of forced draft is used. The blowers, eight in number, are the Sturtevant type, and are located on the protective deck under the breeches portion of the smoke pipes, two for each fireroom. The fans are 69 inches in diameter; width of rim at periphery, 25 inches. The fans are driven by double upright enclosed engines; diameter of cylinders, 7 inches by 5-inch stroke.

ASH HOISTS.

One ventilator in each fireroom is used for hoisting ashes. A Williamson engine, having two cylinders with cranks at right angles, is located in each fireroom-hatch enclosure at the main-deck level and operated by means of hand wheels and geared shafting from the hoisting station. Bells with pulls are fitted for signalling when hoisting ashes. From the hoisting stations the ash buckets are carried by trolley tracks to the ash chutes at the ship's side.

DISTILLING APPARATUS.

The distilling plant consists of four evaporators and six distillers, with their accessories. The evaporator room is located on the berth deck amidship, between Nos. 3 and 4 fireroom-hatch enclosures. The distillers are just above the evaporator room in a trunk at the height of the gun deck. The evaporators have a combined capacity of 21,000 gallons of water per twenty-four hours, and the distillers 20,000 gallons of potable water per twenty-four hours. The evaporators are the horizontal type, with shells of steel plate, with brass tubes 1½-inch iron-pipe size. The tubes and heads are in two sections to facilitate their removal for scaling. The shells are clothed

with hair felt and lagged with galvanized iron. The distillers have cast-iron shells and heads, 5/8-inch brass tubes tinned on both sides. The distiller circulating pump is a Davidson pump having the same dimensions as the hot-well pump. In addition to the distiller connection to the circulating pump, there is a connection from a riser to the fire main. The evaporator feed pump, 5½ inches by 5 inches by 8 inches, draws from the sea and from distiller circulating-water discharge, and discharges into the evaporators. The fresh-water pump has the same dimensions as the evaporator feed pump, and discharges at will into the fresh-water tanks, the main feed tanks, and the reserve feed tanks. The distilled water from the distillers drains into a small tank to which the fresh-water pump has its suction connection.

REFRIGERATING PLANT.

The refrigerating plant is located on the starboard side of the berth deck, the ice-machine room between frames 73 and 75⅓, and the cold-storage rooms between frames 68 and 73. The ice machine is an Allen, horizontal, dense-air type, capable of producing the cooling effect of two tons of ice in twenty-four hours. The cooling pipes are arranged so the air will go direct to the cold-storage rooms, or through the ice-making tank and thence to the cold-storage rooms and scuttle butt, or from the ice-making tank directly to the scuttle butt.

ANCHOR WINDLASS.

A Hyde anchor windlass is located in a pocket between the main and gun decks. A reversible, two-cylinder, vertical, inverted windlass engine with two 15-inch cylinders of 14-inch stroke is located just forward of the windlass pocket on the gun deck.

STEERING ENGINE.

A Hyde two-cylinder, horizontal, direct-acting engine having two 16-inch cylinders of 12-inch stroke is located in

the steering-gear room aft of frame 102 on the upper platform deck, and arranged for steam and hand control. On the starboard guide rod of the steering engine, the rudder angles are calibrated and marked for every five degrees, port and starboard. Rudder hard over to starboard at 35 degrees, 59 minutes, 5.4 seconds; port, 37 degrees, 43 minutes, 57.6 seconds.

The ship can be steered from six stations as follows: by steam from flying bridge, pilot house, conning tower, communication room on upper platform deck forward and from steering-gear room, and by hand from steering room on upper platform deck aft.

HEATING SYSTEM.

Steam is used to heat all living and other spaces requiring heat throughout the ship. The system is divided into two sections, and these sections again into circuits. The forward section has four circuits and heats all parts of the ship forward of frame 57, while the after section has five circuits and heats all parts of the ship aft of frame 57. The steam-distributing manifold for forward circuits is located on the gun deck, between frames 36 and 37, and for after circuits in main-engine hatch above armor gratings. The drains from each circuit lead to drain manifolds which discharge through traps, located in the engine rooms, to feed tanks, auxiliary condensers or to bilge at will.

Steam to galley, pantries, sick bay, bath tubs and shower baths is entirely independent of the heating system. One circuit supplies steam to officers' baths and pantries aft, while another, forward, supplies the forward pantries, baths and sick bay. The drains from these two circuits lead into the auxiliary exhaust. An independent lead supplies steam to the galleys, the drain from which is passed through a separate trap which discharges at will in the feed tank, auxiliary condenser or bilge.

ENGINEERS' WORKSHOP.

The engineers' workshop is located on the port side of the berth deck abreast of the engine-hatch enclosure, and the following tools are provided: One screw-cutting back-geared lathe, swinging 30 inches over the ways and taking 10 feet between centers; one 14-inch screw-cutting back-geared lathe, taking 4 feet between centers; one column-shaping machine of 16-inch stroke and 24-inch traverse; one back-geared drilling machine, to drill up to 1½ inches, 18 inches from edge of work; one 16-inch sensitive drill; one No. 1 universal milling machine; one combined hand punch and shears, with 6-inch shear blades, capable of cutting ¾-inch round iron, shearing ⅜-inch steel plate, and punching ⅜-inch holes in ⅜-inch mild-steel plates; one emery grinder with two wheels 12 inches in diameter and 2-inch face; one 30-inch grindstone. Four bench vises are located in the workshop and two in the engine rooms. The above tools are driven by a 10-horsepower electric motor.

ELECTRIC PLANT.

The electric-generating plant consists of three 50-kilowatt and two 100-kilowatt generating sets of 125 volts pressure at the terminals. The engines are of the vertical, cross-compound type, built by the Sturtevant Company, and have for the 50-kilowatt sets, H.P. cylinders 8 inches, L.P. cylinders 14 inches in diameter by 8 inches stroke. For the 100-kilowatt sets, the H.P. cylinders are 10 inches, L.P. cylinders 18 inches in diameter by 10 inches stroke. The dynamos are of the direct-current compound-wound multipolar type, and the wiring is arranged on the two-wire feed system. The generating sets are located in two rooms, one above the other, just forward of the firerooms, and below the protective deck. There are about one thousand electric fixtures, complete, with all necessary incandescent lamps; twelve inclosed arc lamps of about 3 ampères capacity in engine and firerooms; six 30-inch search-

lights; two truck lights; two electric night signalling sets, complete; two diving lanterns, with 150-candlepower incandescent lamps; six portable electric ventilating sets; forty $\frac{1}{12}$-horsepower desk and bracket fans; six $\frac{1}{6}$-horsepower bracket fans; one electric whistle operator; all necessary lighting and power boards and panel boards.

Motors.—The following motors are installed to drive auxiliaries outside of the engine and fireroom spaces.

Use.	No.	H.P. Each.	H.P. Total.	Winding.	Current (amp.)	Location.	Makers.
Ventilation:							
70-in. S.P. blower..	2	...	...	Shunt	64	Berth deck.	Sturtevant.
60-in. S.P. blower..	4	...	...	Shunt	64	Engine room and hold.	Sturtevant.
50-in. S.P. blower..	2	...	...	Shunt	36	Gun and berth deck.	Sturtevant.
45-in. S.P. blower..	3	...	...	Shunt	26	Berth deck.	Sturtevant.
35-in. S.P. blower..	6	...	...	Shunt	15	Berth and gun decks and platforms.	Sturtevant.
No. 4 Monogram....	6	...	...	Shunt	10	Gun and berth decks and upper platforms.	Sturtevant.
No. 3 Monogram....	3	...	...	Shunt	6.8	Gun decks.	Sturtevant.
Boat cranes.....	2	15	30	Series	105	Upper deck.	Thresher.
	4	20	80	Series	140	Upper deck.	Thresher.
	2	30	60	Series	212	Upper deck.	Thresher.
Deck winches.....	7	25	175	Series	180	Main and upper decks.	Sturtevant.
Ammunition hoists.	27	2.5	67.5	Shunt	19	Hold, ammunition passage and berth deck.	Sturtevant.
General workshop..	1	10	10	Shunt	30	General workshop.	Sturtevant.
Fresh-water pumps	2	2	4	Shunt	15.5	Berth deck and upper platform.	Sturtevant.
Sanitary	1	$\frac{1}{4}$	$\frac{1}{4}$	Shunt	6.2	Engine room.	Sturtevant.
Laundry	1	6	6	Shunt	42	Laundry.	Sturtevant.
Bread mixer.....	1	2	2	Shunt	14	Bakery.	
Dishwashing.....	1	...	...	...	...	General mess pantry.	

TRIALS.

Since it was impossible to secure the vessel at the dock and make more than 60 revolutions per minute, it was decided by the contractors to have a builders' trial at sea, off Delaware breakwater, before asking for the Trial Board. The *St. Louis,* therefore, left Philadelphia April 4th, and after adjusting compasses on the morning of the 5th, headed straight out to sea. A series of runs were made at progressive in-

creases of speed for periods of about twenty minutes each. The maximum speed was maintained for twenty-five minutes. One hundred and twenty revolutions per minute was maintained under natural draft. On this trial the main engines and all auxiliaries worked satisfactorily. After the return to Philadelphia the vessel was gotten in readiness for her official trial, leaving on the morning of May 8th for Boston. The following day, on the way up the coast, the engines were run at full speed for one and a half hours. The evening of the 9th anchored inside of Boston Light Ship, and went into dry dock at Boston Navy Yard at 12 noon, the 10th. Left the dry dock the 12th, and proceeded to Rockland, Me., coming to anchor Sunday morning, the 13th.

The standardization trials were run on the 16th, beginning at 6·52·10.1 and ending at 9·54·52.5 A. M. The results of these runs are shown in Tables I and II, from which the curves on Plate I were platted.

For the high runs picked Pocahontas coal was used, and ordinary run-of-the-mine coal for the lower runs.

The speed curve for these runs showed that 149.6 revolutions per minute would be required to make the contract speed of 22 knots per hour.

At 9·21 A. M., May 17th, the four-hours' endurance run was begun under excellent weather conditions which continued during the trial. The average speed for the four hours was 22.13 knots, with an average of 150.78 revolutions per minute. The main engines ran smoothly and without the use of water on any of the bearings or working parts.

Before leaving the contractors' yard the valve gear of the H.P. cylinders was changed to cut off at 85 per cent. of the stroke. No live steam was used in the receivers at any time during the trial. Loss of fresh water during the four-hours' trial was 17.22 tons. A record was no doubt made in the small amount of oil used. During the twelve days' absence from Philadelphia, including the trials, only 1,982 gallons were used. The oil used was supplied by the Vacuum Oil Company.

Table I.—U. S. S. *ST. LOUIS.*

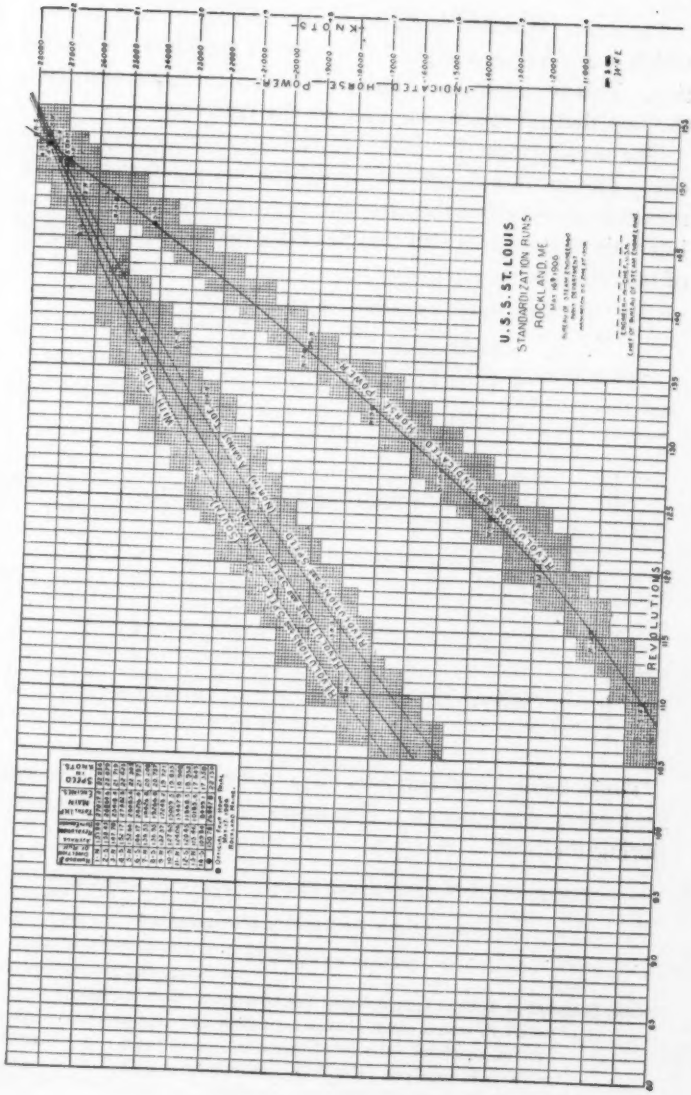
Data from Standardization Trial over Measured Course off Rockland, Me., May 16, 1906.

Number and direction of run.	Time on course.	Revolutions of main engines.					Speed in knots per hour.	Indicated horsepower.											
		To make 1 knot.		Per minute.				Starboard engine.					Port engine.					Total of both main engines.	Auxiliaries in operation.	Total for all machinery.
		Starboard.	Port.	Starboard.	Port.	Mean.		H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.	H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.			
	H. M. S.																			
1, N.	2-41.9	410.65	407.45	152 43	151.24	151.84	22.236	4,023.8	3,818 1	2,830.6	2,805.0	13,477 5	4,026.5	3,883.0	2,808.6	2,821.4	13,539.5	27,017.0	368.4	27,385.4
2, S.	2-43.05	410.20	407.60	150 90	149.95	150 43	22.079	4,017.5	3,638.1	2,776.8	2,802.2	13,234.6	4,026.1	4,037.7	2,784.6	2,784.6	13,630.0	26,864.6	449.2	27,313 8
3, N.	2-45.75	409.30	405.00	148.57	147 01	147 79	21.719	3,787.8	3.535.3	2,721.4	2,784.0	12,828.5	3,582.2	3,820 5	2,630.8	2,556.4	12,589.9	25,418.4	485 9	25,904.3
4, S.	2-40.55	408.50	405 05	152.81	151.52	152.17	22.423	3,889.0	3,827.6	2,876.4	3,057.0	13,650.0	3,931.5	4,127.5	2,941.7	2,852.1	13,852.8	27,502.8	475.4	27,978.2
5, N.	2-41.40	412.25	409.05	153.25	152.06	152.66	22.305	4,026 9	3,933.4	2,974.3	3,155.3	14,089.9	4,048.0	4,094 5	3,054.9	2,798.1	13,995.5	28,085 4	473 0	28,558 4
6, S.	2-45.20	402.60	401.30	146.40	145.93	146.17	21.792	3,468.2	3,346.1	2,496.3	2,644.5	11,955.1	3,457.2	3,692 0	2,623.8	2,488.3	12,261.3	24,216.4	462.6	24 617.0
7, N.	2-58.15	409 70	401.70	137.87	135.18	136.53	20.208	3,141.7	2,823.2	1,978.4	2,187.9	10,131 2	2,897.5	2,666.6	1,905.7	1,825.8	9,295.6	19,426.8	381.6	19,745.7
8, S.	2-53.6	401.05	391.40	138 61	135.28	136.95	20.737	3,002 2	2,864.3	2,106.0	2,199.6	10,172.1	2,777.5	2,651.6	1,884.2	1,781.4	9,094.7	19,266.8	351.0	19,556.8
9, N.	3-02.55	401.05	403.95	131.89	132.84	132.37	19.721	2,559.1	2,229.9	1,658.8	1,736.7	8,184.5	2,847.3	2,661 9	1,827.7	1,726.8	9,063.7	17,248.2	318.9	17,502.5
10, S.	3-01.50	387 40	384 85	128.07	127.23	127 65	19 835	2,485.0	1,972.9	1,416.2	1,545.9	7,420.0	2,353.9	2,207.0	1,535.8	1,492.8	7,589.5	15,009.5	335 1	15,280.3
11, N.	3-10.40	392.90	394.45	123.81	124.31	124.06	18.908	2,151.0	1,705.7	1,264.6	1,379.5	6,500.8	2,285.9	2,008.4	1,332.6	1,280.2	6,907.1	13,407.9	311 2	13,656.1
12, S.	3-09.95	380 75	380.95	120.40	120.46	120.43	18 952	1,874.4	1,583.3	1,138.3	1,158.6	5,754.6	2,106.4	1,750.1	1,138.9	1,118.5	6,113.9	11,868.5	297.1	12,102.1
13, N.	3-24.05	393.55	391.70	115.73	115.18	115.46	17.643	1,644.9	1,340.7	976.9	1,025.7	4,988.2	1,884.1	1,514 7	894.4	904.2	5,197.4	10,185.6	291.5	10,416.8
14, S.	3-27.4	380.85	378 50	110.18	109.50	109.84	17.358	1,342.4	1,110 8	874.2	883.5	4,210.9	1,457.6	1,268.5	785.7	776.4	4,288.2	8,499.1	270.5	8,708 9

Table II.—U. S. S. *ST. LOUIS.*

Data from Standardization Trial over Measured Course, off Rockland, Me., May 16, 1906.

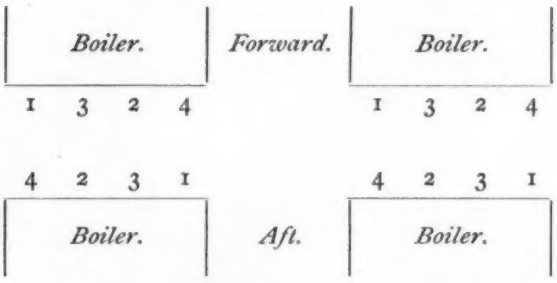
Number and direction of run.	Which engine.	Steam pressures.			Mean effective pressures.					Air pressure, in inches of water.	Cut-off of cylinders from beginning.			Temperatures.						Revolutions or double strokes.						
		In boilers above atmosphere.	At engine above atmosphere.	H.P. steam chest, absolute.	H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Referred to L.P.	Vacuum, in inches.	H.P.	I.P.	L.P.	Outside air.	Engine-room working platform.	Fireroom working level	Injection.	Discharge.	Hotwell.	Feed.	Circulating pump.	Main feed pumps.	Bilge pumps.	Main air pumps.	Auxiliary condenser pump.	Fireroom blowers.	Sanitary pump.	Ampères, Dynamo.
1, N.	S.	269	255	260	117	40	22	21.8	52.38	27	1.50	Full.	Full.	47	78	98	48	110	80	90	134	19.5	84	13	14	320	30	280
	P.		260	265	118	41	22	22.1	53.03	27.5					73		48	98	82	160	144	25.5		18				
2, S.	S.	271	245	260	118	38.5	21.8	22	51.95	27.5	1.56			48	76	100	46	114	84	133	154	26.5	87	16	24	340	30	245
	P.		270	255	119	38	22	22	53.85	27.5					76		46	92	70	136	192			24				
3, N.	S.	261	240	250	119	38.5	21.7	22.8	53.15	27.5	1.60			48	76	98	46	112	86	134	156	32	36	15	48	339	30	230
	P.		240	245	108	41.5	21.8	20.6	50.15	27.5					67		46	92	70	126	192	30		25				
4, S.	S.	275	250	260	112.8	40.3	22.3	23.7	54.71	27.5	1.52			48	76	98	46	114	84	140	148	24.5	42	17	54	359	30	230
	P.		255	255	115	43.5	23	24.3	54.16	27.5					67		46	94	70	128	192	28		22				
5, N.	S.	277	255	265	115.5	41	23	24.8	54.48	27.5	1.68			48	77	99	46	120	90	145	146	29	42	17	31	358	31	230
	P.		255	260	118	42	23.8	21.6	54.39	27.5					75		45	100	88	146	150	35		18				
6, S.	S.	276	250	240	105	38.5	21.3	20.4	49.37	27.5	1.50			50	75	99	45	90	72	124	192	27	44	18	38	318	31	230
	P.		260	240	103	40.4	21.3	18.8	49.77	27.5					76		45	96	80	110	192	27		23				
7, N.	S.	274	265	250	101	38.7	20.7	16.	43.53	27.5	1.00			50	77	101	45	90	77	130	192	31	40	18	28	265	28	240
	P.		270	250	95	31.5	18	18.8	42.73	27.5					77		45	90	77	109	154	23.5		18				
8, S.	S.	260	250	250	96	33	16.5	15.6	43.47	27.5	0.80			52	77	103	45	76	64	118	192	24	22	24	25	250	29	240
	P.		260	255	91	31.3	16.9	15.6	38.56	27.5					76		45	78	74	109	154	17		20				
9, N.	S.	276	260	260	98	37	16.3	15.4	40.48	27.5	0.80			52	76	102	45	75	63	110	192	20.5	34	24	24	229	30	250
	P.		267	263	95	32.6	14.6	13.1	34.31	27.5					79		45	76	74	110	192	20.5		18				
10, S.	P.	261	250	275	86	27.7	14.3	13.9	35.33	27.5	0.60			56	78	107	45	76	63	104	192	19.5	32	24	20	200	30	250
	S.		255	265	88	28	14.1	13.2	31.10	27.5					79		45	74	74	104	148			20				
11, N.	P.	273	265	265	96	31.5	18.2	18.2	38.91	27.5	0.35			56	79	107	45	76	69	134	192	19.5	30	24	22	178	26	250
	S.		265	260	94	30	16.7	17.2	38.31	27.5					79		45	74	69	98	150	13		20				
12, S.	P.	278	275	275	144	50	21	21.2	58.31	27.5	0.40			57	74	109	45	66	68	119	192	14.5	30	24	22	176	28	250
	S.		278	275	135	47.5	23.8	21.2	30.07	27.5					80		45	68	78	120	150	13		24				
13, N.	P.	267	260	250	130	45	18.5	10.5	25.53	27.5	0.35			57	74	108	45	62	65	124	192	14	24	24	22	166	26	240
	S.		265	255	139	47	19	9.2	26.73	27.5					81		45	65	76	88	146	14		18				
14, S.	P.	264	260	255	115	40	17	9.4	22.64	27.5	—			58	74	110	45	62	65	124	192	7	27	24	22	117	26	240
	S.		255	260	122	41	17	8.4	23.20	27.5					74		45	62	65	124	192			24				


U.S.S. ST. LOUIS
STANDARDIZATION RUNS
ROCKLAND, ME
INDICATED HORSE POWER
KNOTS
REVOLUTIONS
SPEED IN KNOTS

Particular attention is called to the amount of coal burned, which was 1.64 pounds per I.H.P. for all machinery in operation. This no doubt was attained by excellent firing and from the fact that no steam was used in the receivers.

The arrangement for firing of furnaces was a very simple one. The furnace doors were numbered as per sketch, and a man was stationed in the after fire room with a watch, and at intervals of one and a half minutes rang electric bells, the cycle of operation being as follows:

1 bell rake No. 1, fire No. 3.
2 bells rake No. 2, fire No. 4.
3 bells rake No. 3, fire No. 1.
4 bells rake No. 4, fire No. 2.



There being four furnaces to a boiler, each furnace was fired every six minutes. The safety valves lifted repeatedly during the trial.

FOUR-HOURS' OFFICIAL TRIAL.—PERFORMANCE.

(*Average of one half-hourly observations.*)

STEAM PRESSURES.

	Starboard.	*Port.*
Steam pressure at boilers (per gauge), pounds.....	266.0	
H.P. steam chest (gauge), pounds....	235.3	238.2
1st receiver (absolute), pounds.....	103.6	108.8
F.L.P. receiver (absolute), pounds...	44.1	43.0
A.L.P. receiver (absolute), pounds...	44.1	43.0
Vacuum in condensers, inches of mercury.....	26.8	26.7

Revolutions, or double stroke, per minute.

Main engines (mean)	151.36	150.2
average for both	150.78	
Pumps, main air	20.5	24.0
circulating	185.7	188.2
feed	28.1	30.8
fire and bilge	33.3	33.4
sanitary	16.9	
auxiliary condenser	0.0	29.5
Blower engines (average)	348.0	
Speed of ship, in knots per hour	22.13	
Slip of propellers, in per cent. of their own speed based on mean pitch	22.84	23.67
Air pressure, in inches of water, average	1.63	

Mean Effective Pressures in Cylinders, in pounds per square inch.

Main engines, H.P. cylinder	111.6	112.3
I.P. cylinder	40.0	43.2
F.L.P. cylinder	21.8	23.6
A.L.P. cylinder	22.45	22.3
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	51.87	54.0

INDICATED HORSEPOWER.

	Starboard.	*Port.*
Main engines, H.P. cylinder	3,810	3,805.9
I.P. cylinder	3,791	4,061.9
F.L.P. cylinder	2,785	2,992.3
A.L.P. cylinder	2,868	2,883.7
Total	13,254	13,693.8
Total, both main engines	**26,947.8**	

AUXILIARIES.

Air pumps, main	10.7	12.4
Circulating pumps, main	73.2	76.0
Feed pumps, main	67.2	77.0
Fire and bilge pumps	3.8	3.9
Sanitary pump	1.5	
Auxiliary condenser pumps	0.0	3.2
Blower engines, forced draft (eight)	122.4	
Dynamo engine (one in operation)	85.4	
Total all auxiliaries	**536.7**	
Main engines, air, circulating and feed pumps	**27,264.3**	
Total all machinery in operation	**27,484.5**	

COAL.

Kind and quality	Pocahontas, hand picked, excellent.
Pounds per hour	44,992

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	19.63
heating surface	0.4294
Main engines, air, circulating and feed pumps per square foot of grate surface	19.47
Main engines, air, circulating, and feed pumps per square foot of heating surface	0.426
Pounds of coal per I.H.P. per hour, main engines, air, circulating and feed pumps	1.65
Pounds of coal per I.H.P. per hour, all machinery in operation	1.64
Pounds of coal per square foot of grate surface per hour	32.4
heating surface per hour	0.703
Cooling surface main condensers, square feet per I.H.P.	0.97
Heating surface, square feet per I.H.P.	2.32

TEMPERATURES.

Temperature, atmosphere, degrees Fahrenheit, average	53.8
engine room, upper platform, average	80.1
working platform, average	72.2
firerooms, above gratings, average	88.0
working level, average	98.0
injection water, average	47.9
overboard delivery, average	100.6
hot well, average	67.7
main feed, average	130.3
smoke pipes, base, average	685.0

SOME ACCIDENTS, REPAIRS, ETC., TO THE VESSELS OF THE TORPEDO-BOAT FLOTILLA (NOVEMBER, 1901, TO JANUARY, 1903), AND OF THE FIRST TORPEDO FLOTILLA (JANUARY, 1903, TO APRIL, 1904). *

By Lieut. Commander L. H. Chandler, U. S. N., Member.

During a service of about two years' more or less active cruising in torpedo vessels of the above flotillas, covering in all a distance of about thirty thousand miles, a number of interesting situations developed as to damages, repairs, etc., which may be of interest to members of the American Society of Naval Engineers, and it is the object of this paper to give some brief account of the more unusual of these. Some of the accidents and repairs involved hulls and not machinery, but they are all presented here in hopes that they will be found of general interest. No attempt has been made to form a continuous and complete story of the cruise in this paper, but the several items are merely presented in fragmentary form, and only the most interesting occurrences have been selected from a mass of such matter, for troubles of the kind were of frequent occurrence, and any attempt to cover them all would result in a paper of almost unlimited length.

* In preparing this account I have drawn freely, and in many cases verbatim, upon the reports of the officers concerned in the work. My thanks are therefore due to the officers of the flotillas, not only for their work, but for the use made of their reports. In connection with the partial docking of and the repairs to the bow of the *Bagley* at the Port Royal Naval Station, I am similarly indebted to Chief Carpenter Henry Rigby, U. S. N., at that time Head of the Department of Construction and Repair at that Station, to whom the success of the operation was largely due.

In more than one instance the character of the work done, and the fact that it was often done in the face of many difficulties and at times when the services of the boats were urgently needed and when failures would have meant serious crippling of the flotilla as a whole, brought letters of commendation to the officers immediately concerned from senior officers present, commanders-in-chief and from the Navy Department.

For a description of the several vessels mentioned the reader is referred to preceding numbers of the Journal of the Society, and to the various official publications covering such matters.

For the composition of the several flotillas, reports of coal consumption, performance at sea, distances steamed, itineraries, etc., reference is made to a paper on the subject published in the last Journal of the "Proceedings of the Society of Naval Architects and Marine Engineers."

BAGLEY—REPAIRS TO BOW. PLATES NOS. 1 AND 2.

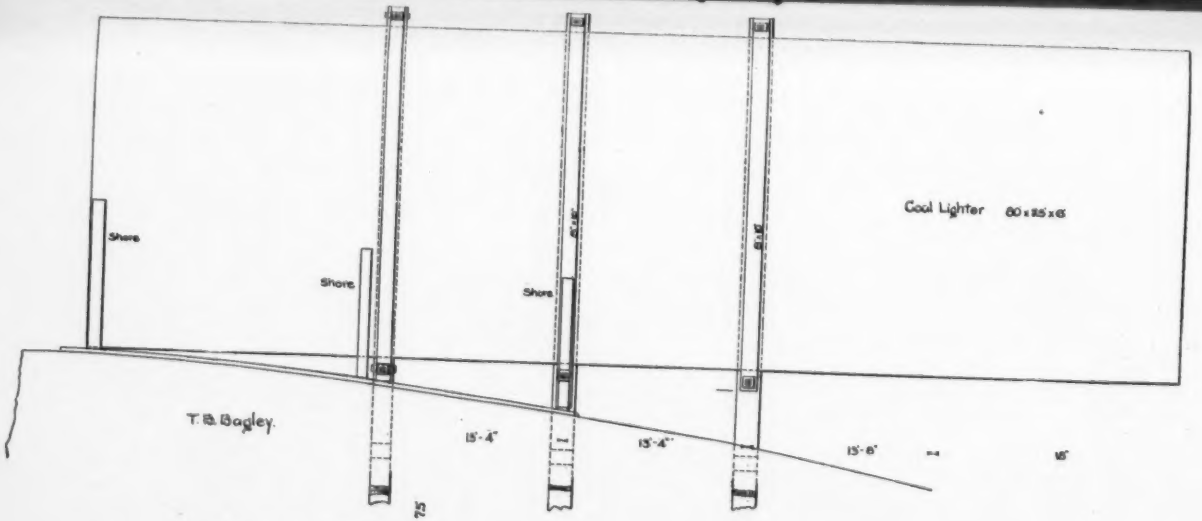
The *Bagley* grounded on her way to Port Royal in the fall of 1901, and a part of the damage sustained was a dent in the bow on the port side, near the keel, resulting in leaky rivets sufficient in extent to cause the bow compartment to fill in a very short time after pumping out. It was determined to repair this at the Port Royal Naval Station, to avoid sending the boat to Norfolk to dock, although the facilities at the former place were not great. There was no dry dock available at Port Royal, the one there being flooded and in use as a wet basin for torpedo vessels in reserve.

The main question involved was the lifting of the bow until the keel came above the surface of the water, without crushing the frail plating or straining the fragile hull.

Two coal barges, rectangular in shape, each 80 feet long, 25 feet broad and 6 feet in depth from the top of the deck to the under side of the bottom planking, were used. In each barge there were two fore-and-aft bulkheads extending from end to end, with limber holes at intervals.

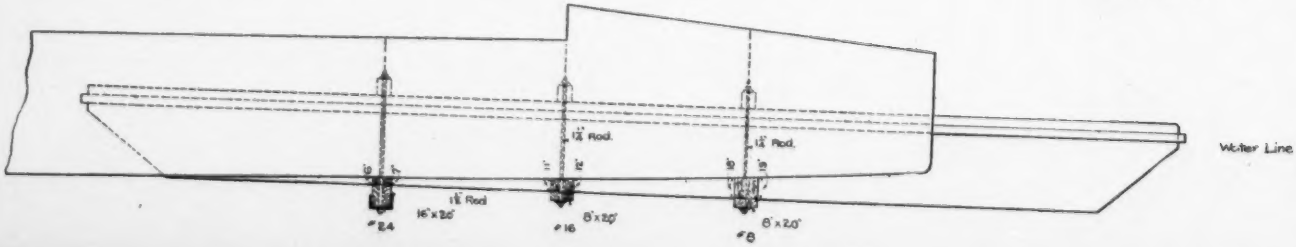
A number of yellow pine timbers 30 feet by 10 inches by 8 inches were used, bolted together in proper lengths, the butts being reinforced on the proper sides by 2-inch plank spiked to the timbers. Three sections in all were used, each being 75 feet by 20 inches by 8 inches when assembled.

These were placed under the bottoms of the barges, while on the tops six sections of timber, each 30 feet by 10 inches by 8 inches were used. Four bar-steel clamps 4 inches by $\frac{3}{4}$


Shore
Shore
Shore
Coal Lighter 80 x 25 x 6'
T. B. Bagley.
15'-4"
15'-4"
15'-6"
15'
75'

60'

Plate No. 1.


1¼" Rod.
1¼" Rod.
1¼" Rod.
16' x 26'
8' x 20'
8' x 20'
#24
#16
#8
Water Line

inch were used, each with a hole punched through the center to take a 1½-inch steel rod. Four steel rods 1½ inches in diameter and 10 feet in length; twelve plate washers ½ inch by 12 inches by 12 inches; eight steel rods 1¼ inches in diameter and 9 feet in length; keel blocks of timber 4 feet by 14 inches; ½-inch chain for preventer purposes, and six short shores with wedges were all used. The bolts were threaded at each end for a length of 6 inches and were provided with nuts. Two ordinary steam siphons were constructed for pumping out the barges and were of 4-inch suction.

The lifting apparatus was assembled by placing the barges so that the rear end of each was abreast the entrance to the dry dock (being inside the flooded dock to get smooth water), they being separated from each other by a space of 17 feet. At 21 feet from the rear end of the barges the first girder was placed. This was composed of two of the 75 feet by 20-inch by 8-inch sections, and was arranged so that the depth was 20 inches. Battens 1⅝ inches in thickness were placed between these sections, thus permitting the rods to pass through and making a width of 17½ inches. The clamps were driven over the bottoms of the two sections for a distance of 10 inches, so that these sections were secured at four points corresponding to the edges of the barges. In the center of the length of this section a keel block was secured which measured 4 feet (length) by 14 inches (width) by 7 inches (heighth forward side) by 6 inches (height after side). Then 1½-inch rods were secured through the clamps by nuts and brought up to the 30-foot by 10-inch by 8-inch timbers which lay across the decks of the barges directly over and parallel to the lower girder, and secured through their ends by nuts set up on plate washers. Half-inch chain was then passed around the timbers as a preventer. At a point 13 feet forward of this station a second girder was similarly rigged, and at another point 13 feet forward of the second still a third was secured. Each of these girders consisted of one assembled timber as described, 75 feet by 20

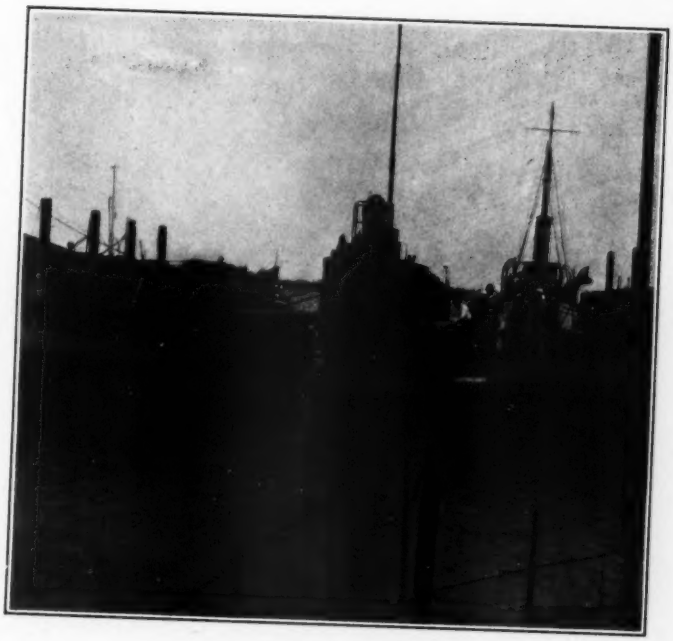


Plate No. 2.

inches by 8 inches, and when in place under the bottoms of the barges had a depth of 8 inches and a width of 20 inches. These girders were secured under and firmly against the bottoms of the barges in a manner similar to that already described for the first girder. The keel block of the second girder was 4 feet by 14 inches by 12 inches by 11 inches, and that on the third girder was 4 feet by 14 inches by 19 inches by 18 inches.

The whole formed a fairly rigid truss, the arrangement being based on the necessity of keeping clear of the sides of the torpedo boat, on the limited space available for working in the flooded dock, on the distance necessary between the barges to take the boat, on the careful elimination of anything that would tend to strain the boat, and on the necessity of getting the maximum lifting power possible out of the two barges.

When in use, the bow of the boat was between the two barges, 18 feet 2 inches and abaft their forward ends, and was supported by the forward keel block under frame number 8, which is 13 feet 6 inches from the forward perpendicular; by the second keel block under frame number 16, which is 26 feet 6 inches from the forward perpendicular; and by the third keel block under frame number 24, which is 39 feet 6 inches from the forward perpendicular. All these frames had transverse watertight bulkheads.

The barges were then filled with water by the pumps of the yard power plant and of the *Bagley* herself, and the boat, which had meantime been trimmed down by the stern as much as possible by shifting weights, was then hauled in between the floats and centered over the keel blocks, and the siphons were started, the yard plant furnishing steam for one barge and the boat herself for the other.

The bow of the boat was thus lifted 2 feet 2½ inches, bringing the damaged plates above water from just below the upper edge of the lower strake of plating.

The barges showed a considerable tendency to heel in towards the boat, the timber girders bending appreciably, so the barges were counterweighted at their outer edges.

The damage was then repaired, and water was run into the bow compartment high enough to test the work, after which the lighters were again filled and the boat floated.

The work as done was eminently satisfactory, the only improvement that suggested itself being that the limber holes in the fore-and-aft bulkheads of the barges should have been plugged, thus enabling them to be kept on an even keel by pumping out more water from the compartments nearer the boat than from the outer ones.

The weight lifted by the barges was calculated as about 70 tons.

A simple device was rigged to definitely ascertain when the boat had landed on the blocks. A double-pointed tack was driven into the top of each block at each end, and through these cords were rove, lying transversely along the top of each block, the ends being brought up onto the floats. So long as these cords would render through the tacks it was apparent that the boat had not settled on the blocks, but when she did take of course it jammed the cords. By this means it was determined that each block was doing its work.

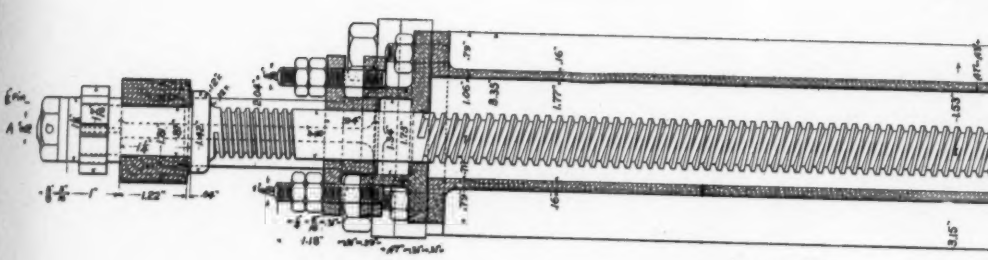
BIDDLE—REPAIRS TO INJECTION PIPE. PLATE NO. 3.

The *Biddle,* while at Port Royal, developed a leak in the main injection pipe of her forward (starboard) engine-room condenser, outboard of the main sea valve. As shown on the plate, this pipe passes through the watertight bulkhead between the engine room and fireroom on the port side, the bulkhead being made up as part of the joint between the sections of the pipe by means of angle irons, etc. In this case certain discrepancies in dimensions had been made up at this point by the introduction of a steel ring about one inch thick into the joint as shown. The whole arrangement as fitted was a very awkward one to handle.

The leak was slight in extent, weeping out between the ring and the bulkhead, principally on the engine-room side. As it indicated rust in a vital bulkhead and in pipe fittings outside the sea valve and in a large pipe, it was deemed necessary to make repairs before going to sea.

ENGINE ROOM.

Tobin Bronz



Section at E.F.

Length of Stem 19.09"

Bulkhead

Tobin Bronze Bolts

FIRE ROOM.

Plate

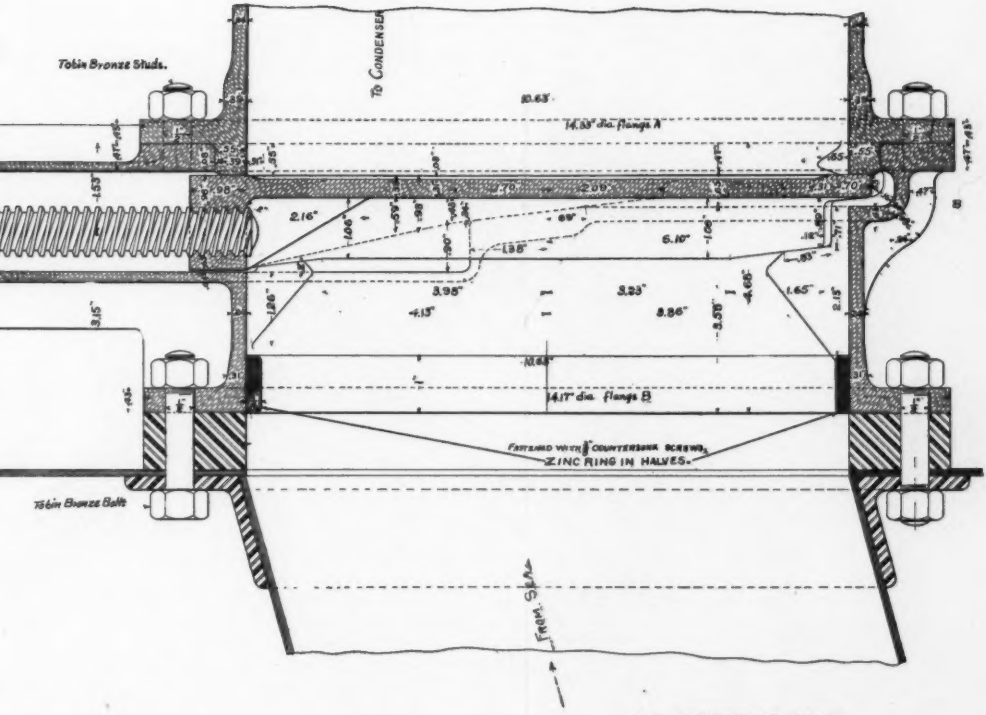


Plate No. 3.

U.S. TORPEDO BOAT BAGLEY.

MAIN INJECTION VALVE.

BUREAU OF STEAM ENGINEERING
NAVY DEPARTMENT
WASHINGTON D.C. APR. 29-1908.

The boat, lying in the flooded dry dock where the water was smooth, was first heeled over to starboard by shifting weights, to the greatest degree possible, in such a way as to bring her starboard quarter down and her port bow up. A diver then went down and plugged the strainer with waste as well as he could, after which a sheet of heavy canvas with a large mass of waste under it was drawn over the inlet and secured by numerous hogging lines and guys. The top of the inlet was from 1½ feet to 2 feet below the surface of the water.

The joint inboard of the valve was then broken and the valve slowly opened, when it was found that the inlet was closed sufficiently tight to make it safe to take out the valve, which was accordingly done.

The bad joint was then broken, and it was found that the bulkhead was intact, but that the faces of the ring were badly corroded. A new ring was fitted, the joint remade, the valve replaced, and the boat righted, after which the inlet pipe was cleared on the outside by a diver.

In both of the above cases the somewhat limited resources of the Port Royal Naval Station were made use of, so that the interest in these two pieces of work lies mainly in the means taken to get at the under-water hull of the boat without the use of a dry dock.

BIDDLE—REPAIRS TO BOW. PLATE NO. 4.

During the search problem off Culebra in December, 1902, the *Biddle* and *Thornton* collided at sea, at night, when running without lights, the *Thornton* being only slightly damaged, while the *Biddle's* injuries were more serious. As shown on the plate, her whole bow was bent around to starboard about 45 degrees, the bend being about 6 feet abaft the stem. The stem piece itself was similarly bent below water, near where it works into the keel. The plating was bent also, that on the port side being in extension, and having one straight vertical bend, while that on the starboard side, being compressed, was crumpled and bent in many curves. The

steel-towing strap on the bow was bent on the starboard side into a "U" shape, while on the port side it remained nearly straight, but came bodily over to starboard with the bow. The doubling up of the plating on the starboard side tore the upper edge of the upper strake away from the forecastle plating. The bull nose was shoved bodily back about a foot and a

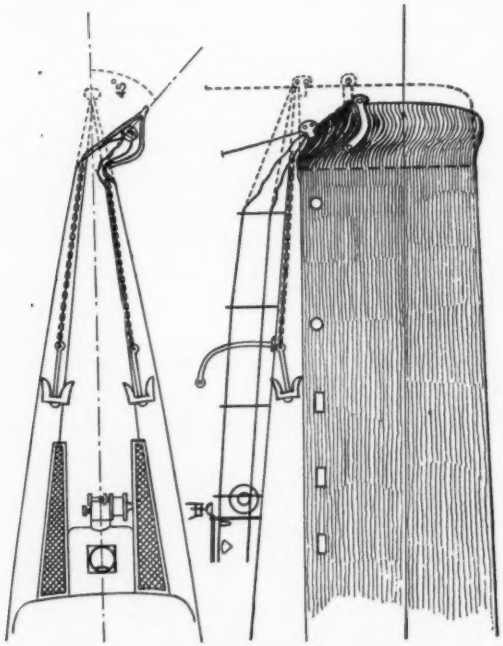


Plate No. 4.

half, crumpling up and tearing the forward forecastle deck plate. On the starboard side the after end of the towing strap was forced in, and the frame of the ship, which is continuous with the deck beam at that point, was bent bodily to port, being shoved in by the bending in of the starboard side of the towing strap. At the draft mark "3" on the starboard side, below the water line, the bow plate had been opened out from the stem for about 1½ inches, the rivets pulling. The

stem piece, consisting of a light piece of steel of "V" cross section, was bent below the water line as above described.

In this condition the boat finished the search problem, running at speeds varying from 14 to 20 knots (63 hours steaming in all) in the open trade sea. The forward compartment was, of course, filled at all times, especially when steaming into the sea, but the collision bulkhead showed no signs of weakness.

To start the repairs a towing shackle was secured in the eye on the steel towing band described above, and the *Biddle* was then towed by it by the *Barney,* using a wire hawser made fast to the shackle. A speed of about 12 to 15 knots was maintained in the ground swell to leeward of Culebra, steaming at the best angles to the sea to get the surges in the proper direction. During the towing the bulge out on the starboard side of the towing strap was beaten heavily with a maul, and inside blocks and wedges were driven in as the towing sprung the bow back into position. After about one hour of this work the bow was sprung back to within less than ten degrees of straight, and the bow plates, which had been sprung together on the inside, had been opened out (the port side from the starboard) enough to admit short jacks between the two sides under the rear ends of the towing strap and at the bent frame. By the use of these jacks and of mauls and wedges, the towing strap and frame were straightened out, the plating coming into shape easily with the strap. This left the bull nose still shoved back, and the plating under and abaft it badly crumpled up. A jack, with blocking from the anchor winch, placed against the rear of the bull nose, soon shoved it forward into place again, and the side plating straightened out very fairly with it, but the forward end of the forecastle plating was badly crumpled and torn. This was cut out and replaced by a $\frac{1}{8}$-inch galvanized-iron plate, cut and fitted to shape. Further judicious use of jacks, wedges, mauls, etc., both inside and out, finally got the bow back into very nearly its original shape.

The bow was then raised by shifting weights aft and by filling the stern compartment with water, until the slight dam-

age below the water could be reached and repaired. Driving a few rivets soon finished the work.

As a result of this work the bow was tight, except for a very slight weeping under the towing strap when steaming into a head sea. The plating and riveting was, of course, somewhat rough, but after it had received a couple of coats of paint it took rather a close inspection to discover that it had been damaged. The boat was thoroughly efficient for service for the rest of the winter.

This work was done entirely by the men of the flotilla, with tools and material (except a few rivets secured from the store ship) carried by the flotilla. There were engaged in the work:

2 machinist's mates, first class, and one oiler..... 10 hours.
4 machinist's mates, first class, and two oilers..... 17 hours.
1 blacksmith ..... 39 hours.

Total ..... 66 hours.

Two days' work, during daylight hours only, with towing for one hour the preceding day, did the work.

BAGLEY—REPAIRS TO BLOWER ENGINE. PLATE NO. 5.

At the end of the winter cruise in the West Indies, and just as the torpedo-boat flotilla was about leaving San Juan for Norfolk, the *Bagley*, by carrying over water, blew off the end of the cylinder of one of her fireroom blower engines. As she has but one blower for each fireroom, this left her with one of her two boilers practically disabled, and in this condition she was certainly not fit to start on her long trip north. It was therefore necessary to either wait a number of weeks at San Juan until a cylinder could be manufactured and shipped down from some navy yard, or else to repair the old one satisfactorily on the spot, which last alternative did not appear practicable at first.

The plate shows the casting as the repairs were finally made. This was a single-cylinder, simple, vertical, inverted engine (shown on the plate upside down), the frame, cylinder and cylindrical valve chest being all cast in one piece, including

the lower heads of the cylinder and valve chest. The larger cylinder, shown to the right in the plate, is the valve chest. The solid ends of the cylinder and chest are shown on top in the plate. The end of the cylinder was blown off, the break

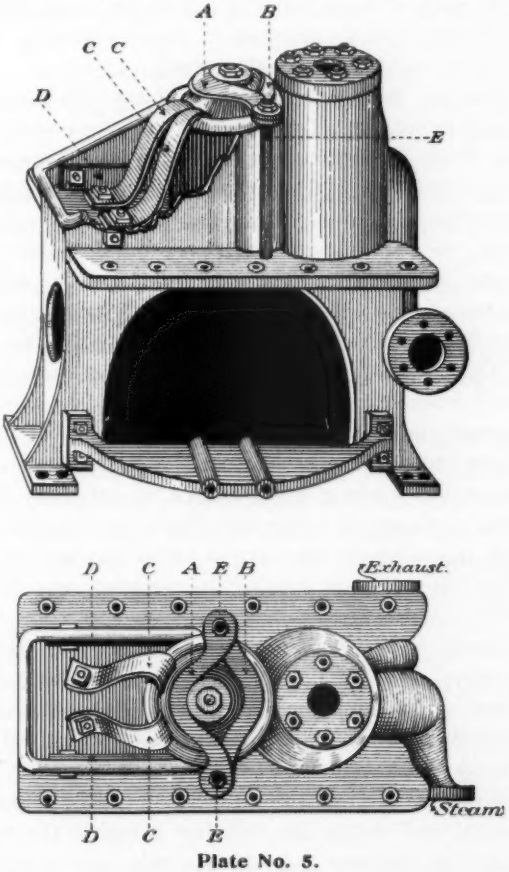


Plate No. 5.

in the casting being circumferential, near the lower end (top in plate), in the counterbore of the cylinder at that end. The whole end of the cylinder came off, carrying with it about a quarter of an inch of the cylinder wall.

Repairs were rendered more difficult by the fact that, in addition to making a tight joint, it was necessary to hold the broken end in place against a tendency to displacement in two directions; it had to be held up against the main body of the cylinder to keep it from blowing off in an axial direction, and it had also to be so strapped in place as to keep it from working off at right angles to the cylinder axis.

The detached part was held from transverse motion by means of the boiler-plate strap marked "D" on the plate, which strap was made to fit around the outside of the cylinder at its lower end, and which was bolted in place by through bolts into the wings of the blower castings as shown.

To hold the end up against the cylinder double sets of straps were used. The first pair, marked "A" and "B", enclosed the swell on the cylinder head as shown, and were drawn up by the bolts "E" extending up through the blower frame. These were reinforced by the two hook-ended clamps "C", which hooked over the edge of the head and set up by bolts through the bed plate as shown.

The joint between the broken parts was made up with red lead, which was given a good chance to set, and a little "smoothon" was spread inside the cylinder to cover the crack, which, as has already been stated, was fortunately in the counterbore and did not involve the wearing surface of the cylinder.

With these rough repairs, which took only a couple of working days, and were done entirely aboard ship, by the flotilla and ship's mechanics, and with ship tools and material only, this blower did its half (and rather more) of the work during the two-thousand-mile run to Norfolk, by way of Key West. It carried without leak the full designed working pressure, and during the last few hours of the run into Norfolk (from Hatteras up), was the only blower the *Bagley* could use, all her coal having been expended from the bunkers of the other fireroom. And these last few hours were at a speed of about 17 or 18 knots as well, so that this final test was a severe one.

DECATUR—HEATING OF BOILER "B".

A case of overheating of boiler tubes occurred in "B" boiler of the *Decatur* on September 5, 1902, while lying alongside the dock at the New York Navy Yard, which is interesting, not so much in itself, as for a number of simultaneous incidents which occurred. Before describing them it may be as well to state that the tubes that had been overheated were rerolled, and that after that the boiler was used for over two years, during which time the ship steamed over twenty thousand miles.

Boilers "A" and "B" were under banked fires for auxiliary purposes, and were both connected through the auxiliary line. Steam was being slowly run up to about 150 pounds for use in making some tests. The only machinery running was the evaporator and the port circulating pump, both in the after engine room, the evaporator being on the after bulkhead, and consequently at practically the opposite end of the steam line from the two boilers.

The machinist on watch at the evaporator first noticed that the oil accumulated on the cylinder end of the circulating pump was burning, and, at about the same time a "T" joint in the steam-supply pipe to the evaporator burnt out. The pressure just before this was noted to be 125 pounds. When the "T" melted, there was a rush of gas out of the pipe, but practically no steam. Had it been steam it would have been difficult for the machinist to close the evaporator valve, which was very near the "T." The machinist immediately shut down everything.

At the same time burning cloth was smelt through all the machinery spaces, and it was seen that the cloth covering to the auxiliary steam pipes was burnt wherever it touched the metal of the pipe, as at the joints, etc.

The man on watch in the live fireroom had the forward main feed pump running, drawing from a tank that had plenty of water in it. The water gauges showed good water, and this man claimed that he had been using the try cocks regularly. This last is probably a misstatement, but the

gauge glasses did show everything as all right. This man noted that the steam suddenly dropped from 125 to 65 pounds and the water suddenly dropped out of sight in the glass on "B" boiler. As nearly as can be determined this happened at the moment when the "T" blew out at the evaporator. Boiler "A" showed no unusual conditions, but it was found shortly afterwards that the automatic auxiliary stop valve on "A" was and had been shut, thus cutting out "A" altogether.

Before an officer or chief machinist could get to the fire-room the men there pumped cold water into the boiler ("B") and started it leaking.

As the steam pressure dropped it was noticed that the manhole gasket in the upper drum of the burnt boiler was smouldering.

The hot boiler was allowed to cool down, and some hours afterwards an investigation was started. Just as men were entering the furnace with a lamp a very pronounced explosion was heard inside the boiler casing, and rusty dust was blown out the funnel and out the furnace doors, the boiler being then practically cool. The men left the furnace, but, after a search for gas inside the casing, again entered. The inspection showed that many of the tubes had been overheated and were leaking, but nothing more.

Just after the explosion the automatic auxiliary-stop valve of boiler "A", which up to that time had resisted all efforts to open it, suddenly opened itself with a loud thump.

No satisfactory explanation of the so-called explosion has ever been found. At first it was thought that it might have been the end of one of the burnt tubes pulling out of the upper drum as it cooled, but, as a matter of fact, it was found that no tube had thus pulled out.

The small balls in the automatic closing gauge glasses were found to be corroded and covered with grease, which accounted for the failure of the glasses to indicate correctly.

The Board ordered to investigate this accident found that it was due to failure of gauge glasses to record and of pump to deliver water to the boiler in sufficient quantity, and, so far

as that feature is concerned, the case was most ordinary. As to effects, however, the Board stated that "the temperature in the steam spaces was raised to at least 800 degrees Fahrenheit, as shown by the fact that the zincs in the top drum were partially melted, but the temperature evidently did not rise much above that (the melting point of zinc is 800 degrees Fahrenheit), for the boiler zincs had only melted down a little in places. The highly superheated steam passed on through the auxiliary steam piping, causing the phenomena stated in the testimony. The temperature on the outside of the top drum and the tubes was lower than that inside, and caused leaks where all tubes are expanded in top drum. Fires were hauled before the boiler was permanently injured."

In addition to the effects already described, the heat over the boiler caused a buckling of the deck across the ship, from side to side. A ridge was raised transversely, about three inches in height at the highest point, and the plating was very hot. This was kept well covered up, so that it would cool gradually, and it returned to place without damaging the hull.

The observed phenomena were as described, and it will be noticed that all of them have not been satisfactorily explained.

DECATUR AND *DALE*—BENT PISTON RODS.

Continuous trouble was experienced with the high-pressure piston rods of these two vessels, more particularly with the *Decatur,* as she was running for about six months before the *Dale* commissioned. It was afterwards learned that the builders of these two destroyers had the same trouble on their trial trips.

These piston rods were hollow, their dimensions being as follows:

Extreme length, inches	58¾
Rod length, inches	44½
Inside diameter, inches	2¾
Outside diameter, inches	4¼
Thickness of wall, inches	1½

The packing originally used with these rods was ordinary Katzenstein's metallic packing.

It will be observed that such a rod has very thin walls for the power to be transmitted, and in addition to this it was found that the builders had already straightened one of the *Decatur's* high-pressure rods, and to do it had taken off a one-sixteenth inch cut, thereby reducing the already small wall thickness.

The troubles experienced were altogether with the high-pressure rods, and it is unnecessary to go separately into the details of the many instances in which these rods heated or bent, it being sufficient to say that it happened day after day, and generally at the most inconvenient times possible. Of course the trouble became worse after each recurrence, and during the winter cruise of the *Decatur* in the West Indies, before the *Dale* commissioned, it became so bad as to greatly cripple the boat.

It should be stated that, from the very first, the utmost vigilence was exercised, and standing orders were in force and were strictly adhered to to stop the engine the instant heating began, and never under any circumstances to put water on a hot rod, the rods, when heated, being always allowed to cool gradually.

It was the concensus of opinion that, with the extremely large hole in the rod, if heating occurred on any one element of the cylindrical surface, there was not enough metal to allow the heat to distribute itself quickly throughout the mass, so that there would be expansion on one side of the rod and not on the other, thus causing a temporary bend in the rod, which in turn immediately caused further heating with a resulting permanent bend.

I have seen a rod do this when a skilled machinist was standing with his hand on the throttle watching for heating, and ready to stop the engine on the first signs of heating; yet even then the rod would heat enough to cause a permanent bend before the engine could be stopped, the rise in temperature being sufficient to burn the oil on the rod. After such a

heating, when running back to port under the other engine, it has been necessary to keep the damaged one from turning over without steam, as the rod would heat up again even at that slow speed of rotation.

As the trouble was progressive, so the steps taken to remedy it became more and more drastic. The form of packing used was changed, first to hard Garlock and afterwards to the softer Selden. Rods were first straightened, and then cuts had to be taken off them to the greatest degree consistent with safety. Finally, in the *Decatur,* one of the low-pressure rods was fitted (after a fashion) in one high-pressure cylinder, the bent rod being put into the low-pressure cylinder, and with this arrangement she finished her winter cruise, speed being reduced to about fifteen knots from that time on.

After the ship returned to the navy yard, new rods of greater thickness were fitted, and a form of metallic spring packing was used throughout. These changes, applied to both *Dale* and *Decatur,* eliminated the trouble.

DALE AND *DECATUR*—HEATING OF STERN TUBES.

The *Dale,* just afer leaving the navy yard, having been docked about a month before, suddenly began to heat both stern-tube bearings, just abaft the stuffing boxes. These bearings were of soft metal, instead of the usual lignum vitæ, and it is thought that sand had gotten into them after docking, while lying in shoal water alongside the navy yard dock. The heating was serious in extent, and even when the stern-tube gland was removed no water would come through.

A small hole was bored in the top of each bearing, and oil, sulphur, graphite, etc., were forced in, the engine being kept in motion all the time at reduced speed, the gland being slacked back and all packing removed. Loose pieces of burnt soft metal began to come out, both through the hole, and into the stuffing box. Continuous running and careful nursing for several days in this way finally got out all the loose metal, secured a flow of water through, and stopped the heating to a large degree, although the bearings were never thoroughly satisfactory thereafter until the shafts had been withdrawn, and new bearings poured.

Upon subsequent docking the bearings in the stern struts were also found to be badly cut. Wood seems preferable to metal for these bearings.

The same trouble was experienced with the *Decatur*, although to a less degree, the experience with the *Dale* serving as a warning.

BARRY—METHOD OF PLUGGING CONDENSER TUBES.

The *Barry*, when steaming in flotilla at moderate speed, developed leaky tubes in both condensers. One engine was speeded up to keep position, and the other was stopped, being allowed to turn over by the drag of the screw. The heads were taken off of the condenser of the idle engine, and the leaky tubes were detected by the use of a burning taper, the air pumps of the dead engine creating enough vacuum in the condenser to make the necessary indraft through the leaky places. These tubes were then plugged, the condenser refitted, the engine started and used to keep position, while the same process was repeated with the other condenser. By this ingenious method the ship kept her position in formation throughout the trouble.

DECATUR—FRACTURED BED PLATES OF MAIN ENGINES. PLATE NO. 6.

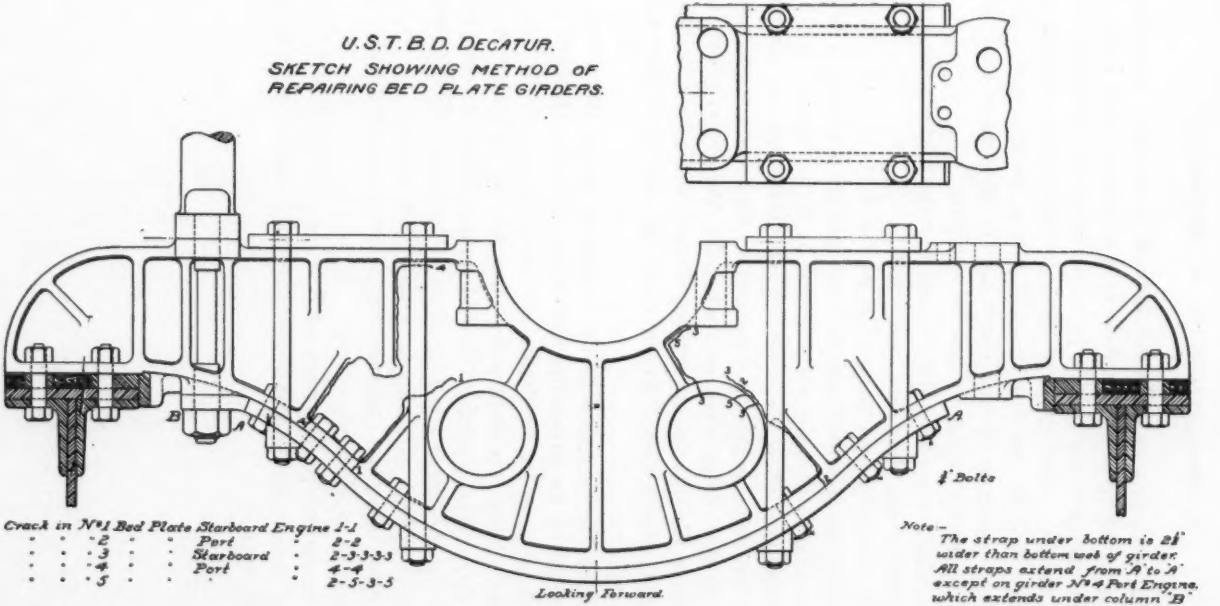
While steaming in the vicinity of Santa Cruz, at about 20 knots, in a long following sea, on November 26, 1902, it was found that cracks had developed in five out of the twelve 'thwartships girders of the main-engine bed plates of the *Decatur*. There were six of these girders in each engine, and three were found to be cracked in one engine and two in the other. The details of these cracks, as well as the method of repairs adopted, are shown on plate.

When the cracks first showed, as the engines seemed to be running smoothly, a set of indicator cards was taken on the port engine before slowing down, which cards showed no abnormal conditions.

The Board order to investigate this damage reported that: "It is the opinion of the Board that these cracks were caused

U.S.T.B.D. DECATUR.

SKETCH SHOWING METHOD OF REPAIRING BED PLATE GIRDERS.



Crack in No 1 Bed Plate Starboard Engine 1-1

"	"	2	" Port	2-2
"	"	3	" Starboard	2-3-3-3-3
"	"	4	" Port	4-4
"	"	5	"	2-3-3-3

Note – The strap under bottom is 2½" wider than bottom web of girder. All straps extend from 'A' to 'A' except on girder No 4 Port Engine, which extends under column 'B'

Plate No 6

by initial strains set up in the material (cast steel) while cooling after being cast; that they were originally what is known as shrinkage cracks which have increased and extended during service. Most of them have the appearance of being old cracks, and in one case, that of No. 4 girder, port engine, there is evidence of an attempt having been made to conceal the original shrinkage crack by caulking."

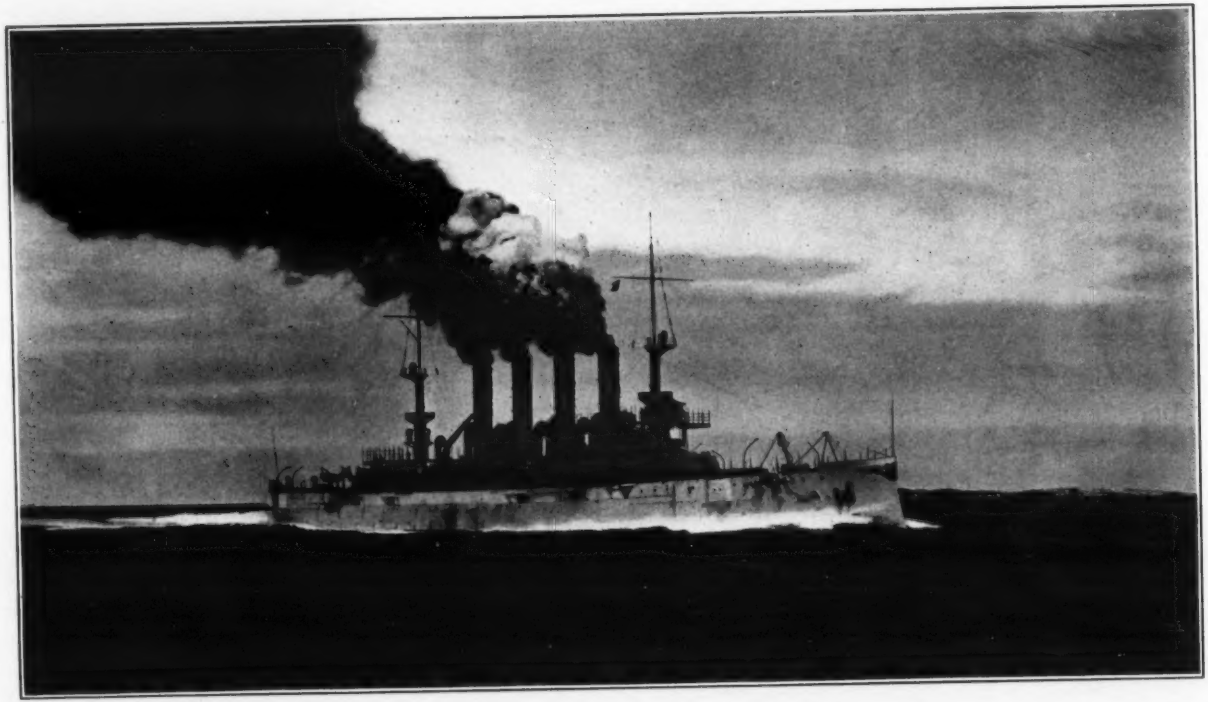
In addition to this it may be said that the fillets in the angles of the castings were very insufficient.

A number of the large ships of the fleet, by order of the Commander-in-Chief, supplied fish plates of half-inch boiler plate bent to fit templates prepared by the carpenter's mate of the flotilla. These were secured over the cracks by machine-finished bolts, as shown in general on the accompanying plate. These fish plates were secured under the girders, the bolts running through the lower webs thereof, while bolts of greater length, outside the girders on each side, ran up to other plates on top.

With the exception of the supplying of the plates bent to template by the large ships and of the use of the lathes of some of those ships by the flotilla mechanics to make bolts, etc., no outside labor was employed on this job, which took from two to three days, working day and night. Upon its completion the *Decatur* made a trial-trip run at 20 knots, and afterwards ran up as high as 22 knots, without any trouble.

When the *Dale* commissioned, several of her girders were found to be similarly cracked.

Permanent repairs were made at the navy yard later by putting in new manganese-bronze girders to replace the damaged steel ones.



U. S. Armored Cruiser "Washington."

DESCRIPTION AND OFFICIAL TRIALS OF THE U. S. S. *WASHINGTON.*

By Wm. A. Leavitt, Jr.

The *Washington* is a twin-screw armored cruiser, and was built by the New York Shipbuilding Company, Camden, N. J.

The ship is one of two authorized by Act of Congress, July 1, 1902; contract for construction was signed February 10, 1903, being one day later than the signing of the contract for her sister ship, the *Tennessee*, built by the Wm. Cramp & Sons Ship and Engine Building Company, of Philadelphia, Pa.

The keel was laid September 23, 1903; the hull launched March 18, 1905, and the ship was ready for delivery July 30, eleven days in advance of the contract date of completion, August 10, 1906 (42 months).

The contract price for the construction of the machinery and hull, exclusive of armor and guns, is $4,035,000.

This is the first naval vessel built by this concern, and that it will be completed in advance of the contract time may be considered a great credit to the builders and a most modern shipbuilding plant.

The propelling machinery was given exhaustive trial at the dock, for a period of five days, in December, 1905; the result being that the usual builders' trial was considered unnecessary, and permission given to run the official trial April 10, 1906.

The ship left Camden March 29, proceeding first to Newport News, Va., for docking, and cleared from the latter place for Rockland, Me., April 7, arriving on the 8th, and making the run in exactly 34 hours, over a course exceeding 645 miles in length, or at an average rate of speed of about 19 knots.

With the exception of two hours, when a spurt was made at full power, the entire run was made under natural draft at about 103 revolutions, and without experiencing trouble of any kind or even the use of the water service.

April 10th being stormy, the standardization runs were made on the 11th over the Government measured-mile course off Monhegan Island, the maximum speed attained being at the rate of 23.293 knots, at 125.96 revolutions; and the average of five runs at full power was 22.546 knots, at 124.97 revolutions. On April the 12th the four-hour endurance run, in open sea, was made most successfully, the average speed being 22.272 knots, at 123.88 revolutions and an average air pressure of 1.3 inches, thereby exceeding the contract speed of 22 knots and keeping well under 1,000 foot piston speed and maximum allowed air pressure of 2½ inches.

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet and inches	502–00
on L.W.L., feet and inches	502–00
over all, feet and inches	504–05½
Beam, extreme, feet and inches	72–10½
at L.W.L., feet and inches	72–10½
Ratio of length to beam	6.9
Depth, molded, main deck, at side, feet and inches	42–02⅜
Draught, normal, mean, feet and inches	25–00
mean, on trial, feet and inches	25–00$\frac{1}{16}$
Displacement, normal draught, tons	14,517
mean, on trial, tons	14,561
per inch on L.W.L., normal draught, tons.	59.72
Area of immersed midship section, normal draught, square feet	1,732.2
L.W.L. plane, normal draught, square feet	25,078
wetted surface, square feet	43,500
Center of gravity of L.W.L. plane, aft of M.P. (1 foot aft of No. 62), feet and inches	8–00
buoyancy above bottom of keel, feet and inches	14–00
aft of M.P. (1 foot aft of No. 62), feet and inches	2–05
Transverse metacenter above center of buoyancy, feet and inches	15–09
Longitudinal metacenter above center of buoyancy, feet and inches	628–06
Coefficient of fineness, block	.55
of M.S.	.955
L.W.L.	.6855
Cylindrical coefficient	.5842
Number of frames	124

MAIN BATTERY.

Four 10-inch B.L.R., 40 calibers in length, in two balanced, electrically operated turrets; sixteen 6-inch R.F. guns, 50 calibers in length, twelve on gun deck and four on main deck; twenty-two 3-inch R.F. guns, 50 calibers in length, twelve on gun deck, ten on main deck.

SECONDARY BATTERY.

Twelve 3-pounder R.F. guns on upper deck; six in hammock berthing, two on forward bridge and four on after bridge; two 3-inch field guns on carriages; two automatic guns, 30 calibers in length, on forward bridge; four 21-inch submerged torpedo tubes, two aft and two forward; two Gatling guns, on upper deck; two 1-pounder semi-automatic guns, on upper deck; two Colt guns for boat and landing purposes.

ARMOR.

All armor face hardened.

Belt armor, 5 inches at top and bottom; protective deck, 3 inches forward and aft, 1½ inches and 4 inches on top and slope; side above belt, 5 inches; athwartship, 5 inches; turret gun, 9-inch front plates, 7-inch sides, 5-inch back, 2½-inch top; barbettes, 7 and 4 inches; 6-inch gun positions, 5 inches; 2-inch splinter bulkheads on gun deck; 2-inch splinter bulkheads on main deck; 2-inch nickel-steel for .3-inch gun positions.

MAIN ENGINES.

There are two main engines, right- and left-hand, outboard turning when going ahead, each in a watertight compartment. They are of the vertical, inverted, direct-acting, 4-cylinder, triple-expansion type, following very closely the Government design. The arrangement of the cylinders, beginning forward, is, forward L.P., H.P., I.P. and after L.P., respectively, and are braced to prevent fore-and-aft and athwartship motion.

The crank shaft is arranged with the cranks 90 degrees apart, and is in two sections, the forward L.P. and H.P. on one and the I.P. and after L.P. on the other, the sequence of cranks being H.P., I.P., forward L.P. and after L.P.

The cylinders are supported by a framing consisting of twelve forged-steel columns, flanged top and bottom for bolting to the bed plates and bottom of cylinders. The framing is trussed by two forged-steel stays symmetrically arranged between each athwartship pair of columns, with fore-and-aft horizontal and diagonal tie rods leading from the middle pairs of columns, thereby offering no resistance to cylinder expansion.

The turning and reversing engines are located on the outboard side of the engines, the former being aft of the H.P. cylinder, with the worm wheel over the shaft-coupling flanges, and the latter forward of the I.P. cylinder. The throttle valve and all handling gear is located on the inboard side between the cylinders mentioned, with the gauge board and other signal details directly back of it on the center-line bulkhead. Piston valves are used on all of the cylinders, and are operated by the Stephenson double-bar link motion, with adjustable cut off. There are five graduations for points of cut off on each reverse-shaft arm, and are for the ahead motion only.

The material for the cylinders is cast iron, fitted with close-grained, hard cast-iron piston and valve-chest liners. The steam-jacket space is between the piston liner and cylinder body. The body of the H.P. piston valve is cast iron, and of the I.P. and L.P. valves, steel pipe with cast-iron heads.

The packing rings, of hard cast-iron, are practically solid, being turned larger than the bore of the valve-chest liners, cut obliquely, the abutting ends bolted together and finished to fit liners. The H.P. valve stem is connected to the link block, and the I.P. and L.P. valve stems to a cast-steel crosshead which is connected to the link block. The crosshead guides, which are of cast-steel, are bolted to the valve-chest covers. The suspension links, reverse shafts, eccentric rods and valve stems are high-grade machinery forgings.

CYLINDER DATA.

	H.P.	I.P.	L.P.
Diameter, inches	38½	63½	74
Stroke, inches	48	48	48
Thickness of body, inches	2½	2	1¼

	H.P.	I.P.	L.P.
Thickness of liner, inches	1⅝	1⅝	1½
jacket space, inch	¾	¾	¾
valve chest, inches	1⅝	1½	1½
Number of studs in cylinder covers	38	58	63
Diameter of studs in cylinder covers, inches	1⅝	1½	1⅝
Pitch of studs in cylinder covers, inches	4⅛	4⅛	4
Number of studs in valve-chest covers	20	24	20
Diameter of studs in valve-chest covers, inches	1½	1½	1½
Pitch of studs in valve-chest covers, inches	4⅛	4⅛	5¼
Number of tap bolts in piston liners	28	42	40
Diameter of tap bolts in piston liners, inch	1	1	1
Pitch of tap bolts in piston liners, inches	4¼	4¼	5¼
Number of piston valves	1	2	2
Diameter of piston valves, top	24½	27	27
bottom	24 7/16	26⅛	26⅛
balance pistons	9	9	9
Port area through valve-chest liner, top, inches	274.78	296.8	296.8
bottom, inches	273.9	295.9	295.9
Valve travel, inches	10	10	10

VALVE DATA FOR PORT ENGINE.

Steam lead, top, linear, inch	1/16	27/32	1⅛
bottom, linear, inches	⅛	63/64	1 1/64
Full gear cut-off, top, per cent	77.0	78.25	78.1
bottom, per cent	70 4	69.0	69.4
Earliest cut-off, top, per cent	50.8	56.9	56.7
bottom, per cent	43.3	46.1	44.2
Diameter of valve stem through gland, inches	3¼	3¼	3¼
valve, inches	2¼	2¼	2¼
Area of valve-stem crosshead guide, each side, square inches	31½	78	78

ASSISTANT CYLINDERS.

One of the features of the propelling machinery is the Lovekin improved assistant cylinder applied to the valve gear of the main engines. As this is the first appearance of these cylinders on a vessel of the United States Navy, their trial performance was naturally watched with considerable interest.

To the practical sea-going engineer, with whom results only are significant, the smooth, quiet running of the valve gears furnished convincing evidence of an agent that was doing its allotted work in a simple yet most effective manner, and to the designer who, in addition to solving this perplexing problem, has had to overcome the inertia of conservatism,

the successful evolution of a device based primarily on theoretical conclusions, it is a complete triumph.

As a brief illustration, there is here given an analytical diagram, Plate I, of the assistant cylinder on the starboard high-pressure valve gear. The ordinates of the liners, A–B and B–C, measured from the base line, give the inertia resistance to be overcome by the cylinder at the various stages of the valve travel. The compressions and expansion curves are exact reproductions of the cards taken on the official four-hour trial. The difference in the lengths of the ordinates of the upper and lower curves gives the ordinate of the resultant curve.

The shaded area between the inertia lines and the resultant effort curve represents the unbalanced work which has to be done by the eccentrics, and is fully explained in the following articles, the limited scope of this article not permitting an exhaustive analysis of the subject in question. Interested readers will be well repaid by consulting the references mentioned:

"Balanced Cylinders Theoretically and Practically Considered," JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XVI, No. 3, and XVII, No. 3.

"The Balancing of Valve Gears," "Journal of the Society of Naval Architects and Marine Engineers," November, 1902.

PISTONS.

They are conical shaped, rough machined all over, the material of the H.P. being cast iron, and that of the I.P. and L.P. cast steel, the followers in each case being of the same material. The pistons are bored taper to fit the rods, and have a square counterbore on the under side to suit the shoulder on the piston rods. The pistons are secured to the rods by $7\frac{3}{4}$-inch composition nuts locked in place by plates bolted to the piston.

The packing rings are of hard cast iron. Those of the H.P. and I.P. pistons are practically solid, having been cut obliquely for machining and the ends clamped solidly together.

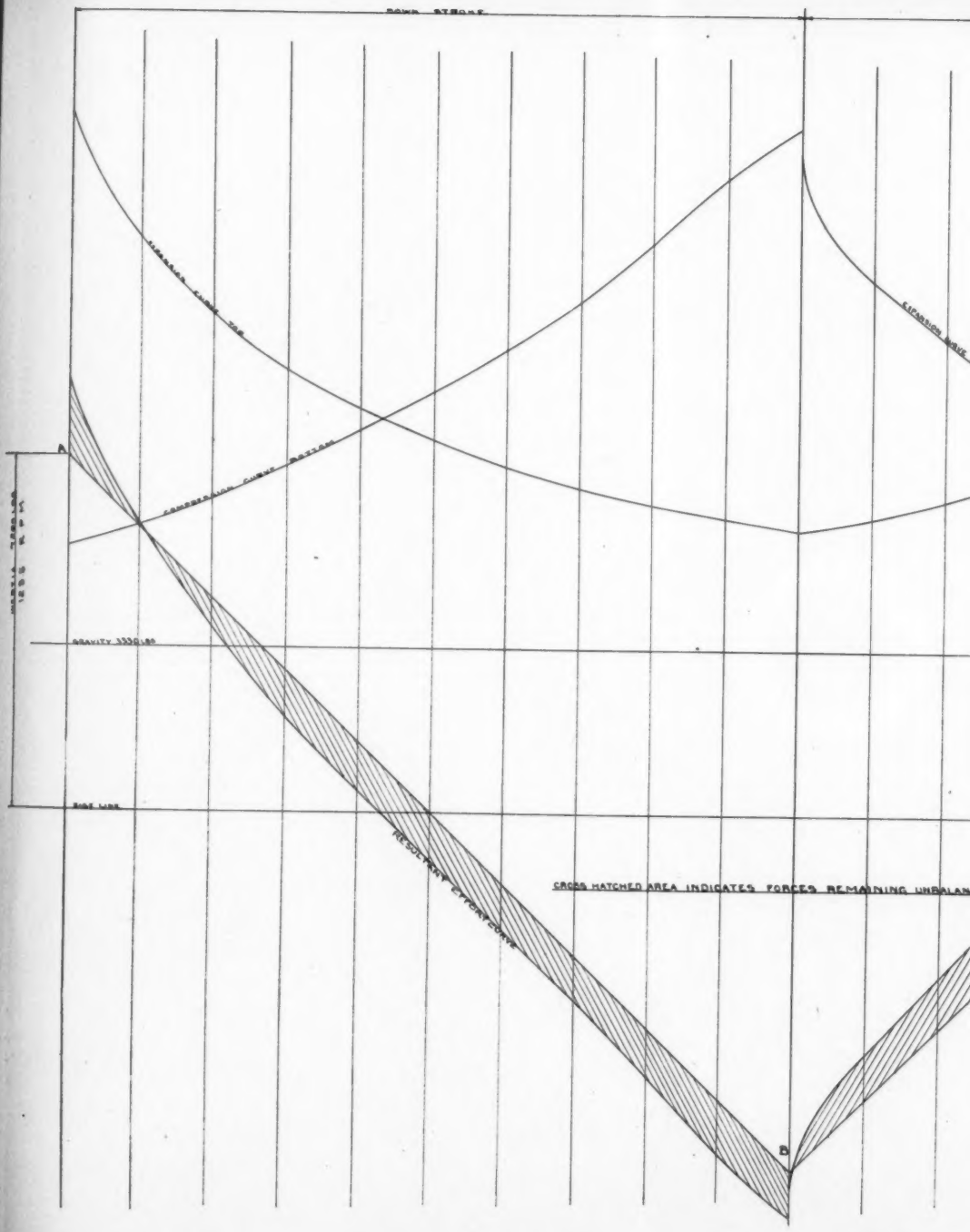
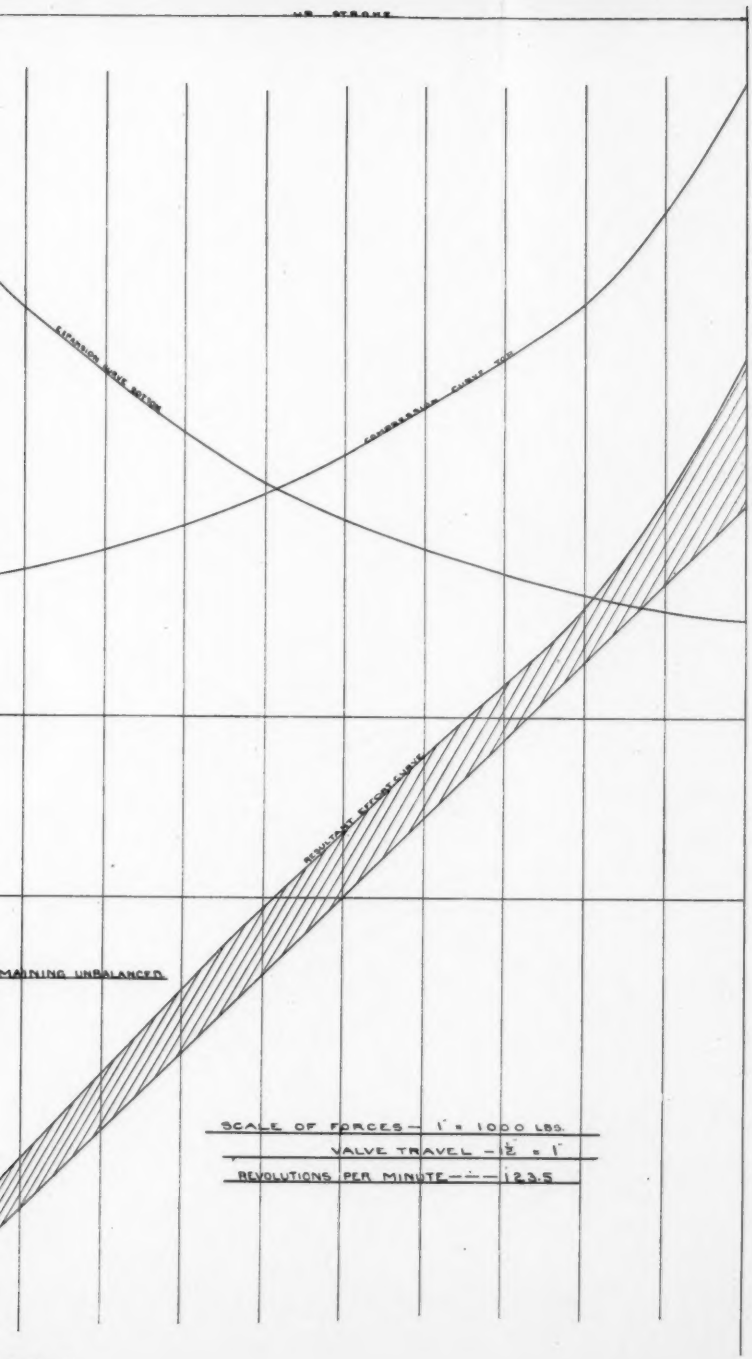

DOWN STROKE
A
B
GRAVITY 333 LBS
RESULTANT EFFORT CURVE
CROSS HATCHED AREA INDICATES FORCES REMAINING UNBALAN

Plate I.



TOP END.

BOTTOM END.

SCALE OF SPRING-1"=125LB.

SCALE OF FORCES— 1" = 1000 LBS.

VALVE TRAVEL — 1/2 = 1"

REVOLUTIONS PER MINUTE — — — 123.5

DIAMETER OF CYLINDER = 9"

TRAVEL OF VALVE = 10"

H.P VALVE GEAR. STARBOARD ENGINE.

ANALYSIS OF LOVEKIN'S IMPROVED ASSISTANT CYLINDER ON U.S. ARMORED CRUISER WASHINGTON FROM CARDS TAKEN ON OFFICIAL FOUR HOUR TRIAL APRIL 12, 1906.

J. D. Lovekin
May 8, 1

Plate I.

Lugs cast on the back of the rings limit the play, the bore being $\frac{1}{8}$ inch greater than the corresponding bore of the piston. Those for the L.P. pistons are in three sections, cut obliquely, tongued and clamped in such a manner as to prevent expansion beyond a given diameter.

Number of packing rings in pistons	1
Width of packing rings in pistons, inches	$2\frac{1}{2}$
Thickness of packing rings in pistons, inch	1
Number of follower studs in H.P. pistons	12
I.P. pistons	17
L.P. pistons	20
Diameter of follower studs, inches	$1\frac{1}{2}$

PISTON RODS, CROSSHEADS AND CROSSHEAD SLIPPERS.

The piston rods are high grade, and the crossheads and slippers Class A, forged steel, the slippers being faced with white metal. The ends of the piston rods are tapered to fit the pistons and crossheads, and the slippers are bolted to the crossheads.

The ahead guides are of cast iron, made hollow for the water service, and the backing guides are of cast steel, bolted to the ahead guides. The guides are of the slipper type, bolted at the top to flanges provided at the bottom of the cylinders and at the bottom to cast-steel strongbacks, carried by the engine framing.

	H.P. and I.P.	L.P.
Diameter of piston rods, inches	$8\frac{1}{2}$	$8\frac{1}{2}$
axial hole, inches	$4\frac{1}{2}$	5
crosshead pins, inches	$10\frac{1}{2}$	$10\frac{1}{2}$
Length of crosshead pins, each, inches	$11\frac{3}{16}$	$11\frac{3}{16}$
slippers, inches	30	30
Width of crosshead slippers, inches	27	$20\frac{1}{2}$
Length of backing surface, inches	30	30
Width of backing surface, each, inches	$9\frac{1}{2}$	$6\frac{1}{2}$

CONNECTING RODS, CRANK SHAFTS AND BED PLATES.

The connecting rods and crank shafts are high-grade machinery forgings. The upper ends of the connecting rods are forked for the crosshead-pin brasses; these brasses and those of the crank pin being lined with white metal. All rods, crank shafts and crank pins are hollow, the holes in the pins being used as oil reservoirs.

The bed plates and main bearing caps are of cast steel, the caps and bottom brasses being hollow for water service and lined with white metal. The bottom brasses are circular to facilitate removal.

	H.P. and I.P.	L.P.
Length of connecting rod, center to center, inches	96	96
Diameter of connecting rod, top, inches	8¼	8¼
bottom, inches	9¼	9¼
hole, body, inches	5	5¼
crosshead end, inches	2	3
pin bolts, inches	3½	3½
Number of crosshead-pin bolts	4	4
Diameter of crank-pin bolts, inches	4¼	4¼
Number of crank-pin bolts	2	2
Diameter of crank pins, inches	20	20
Length of crank pins, inches	23	23
Diameter of hole in crank pins, inches	12	12
shaft, inches		10
crank shaft, inches		19½
journals, inches		18½
coupling flanges, inches		32
Width of crank-shaft coupling flanges, inches		4½
Number of coupling bolts		12
Diameter of coupling bolts, inches		3½
pitch circle, inches		26
Width of crank webs, inches		21½
Thickness of crank webs, inches		12
Number of main bearings		6
Length of main bearings, H.P., inches		22½
I.P., inches	22½ and	31
forward L.P., inches		22½
after L.P., inches	22½ and	31
Diameter of main-bearing bolts, inches		3
Number of main-bearing bolts per bearing		4

REVERSING ENGINE.

The reversing engine is located forward of the center line of the I.P. cylinder, with its steam cylinder bolted to the outboard side of the main-engine bedplate. There is an oil-control cylinder of brass connected to the steam cylinder by wrought-steel stanchions which serve as crosshead guides. The reverse shaft is also on the outboard side of the engines, near the top of the engine framing, and is connected to the crosshead of the reversing engine by two forged-steel connecting rods. Differential levers control the valve motion.

DATA FOR ENGINE.

Diameter of steam cylinder, inches	16
oil cylinder, inches	9
steam piston rod, inches	3½
oil piston rod, inches	3½
tail rod, inches	3½
Stroke, inches	22
Length of crosshead bearing, each, inches	8
Width of crosshead bearing, each side, inches	4
Length of crosshead pin, each, inches	3
Diameter of crosshead pin, each, inches	3½
tie rods between cylinders at crosshead bearing, inches	3½
between cylinders at ends, inches	3
connecting rods, ends, inches	2½
middle, inches	3
Length of connecting rods, feet and inches	4–0 3 31/32
reverse levers, inches	18⅞
Diameter of reverse shaft, inches	8½, 8, 7½
bearings, inches	8, 8, 7
Length of reverse-shaft bearings, inches	8½

TURNING ENGINE.

The turning engine is located on the main-engine bedplate, on the outboard side, aft of the H.P. cylinder. The shaft carries a worm that engages a worm wheel carried by an inclined shaft, at the lower end of which is another worm that engages the driving worm wheel located on the crank-shaft coupling flanges. The second worm is keyed to the inclined shaft by means of a feather, but may be moved vertically to engage or disengage the worm wheel on the crank shaft. A square for a ratchet, for turning by hand, has been provided on the after end of the turning-engine shaft.

The time required to turn the main-engine crank shaft through one revolution is about three minutes.

DATA FOR TURNING ENGINE.

Number of cylinders	2
Diameter of cylinders, inches	6
piston rods, inches	1½
valves, inches	3½
Stroke, inches	7
Valve travel, inches	1½
Crank angle, degrees	90
Area of crosshead slipper, inches	16⅝

Diameter of crosshead pin, inches	1¾
Length of crosshead pin, inches	2¾
Diameter of crank pin, inches	2½
Length of crank pin, inches	2½
Diameter of crank shaft, inches	2½
Length of crank-shaft bearings, inches	4 and 3
Diameter of pitch circle of worm, inches	6 1/8
wheel, inches	17 3/8
Number of teeth	36
Diameter of pitch circle of vertical worm, inches	9
worm wheel, inches	64 5/8
Number of teeth	60
Diameter of vertical shaft, inches	4¼
Area of step bearing, square inches	38½

SHAFTING AND THRUST BEARING.

The shafting is arranged in three sections, as follows: thrust, stern-tube and propeller, all of Class A, forged steel. The thrust and stern-tube shafts are coupled as follows: A forged-steel collar fits on the end of the stern-tube shaft, secured to prevent turning by three keys, and into a circumferential groove in the end of the shaft is fitted a ring made in halves, the coupling bolts passing through the steel collar, the half rings and the flange on the thrust shaft. The stern-tube and propeller shafts are coupled as follows: Over the tapered ends of the shafts is fitted a long sleeve bored with a double taper to fit that of shafts and secured to each by two keys and a taper cotter.

The thrust bearing is of the horseshoe type, with a steady bearing at each end, fitted with glands to prevent the escape of oil. The body is of cast iron, so shaped as to form an oil reservoir, and the horseshoes are of cast steel, made hollow for the oil and water service. They are faced with white metal and have oil grooves on the bearing surface.

DATA FOR SHAFTING.

Diameter of thrust shaft, inches	17¼ and 17½
hole, inches	10½
Number of thrust-shaft collars	12
Outside diameter of thrust-shaft collars, inches	27
Width of thrust-shaft collars, inches	2
space of thrust-shaft collars, inches	4
Length of thrust-shaft bearings, inches	18

Number of thrust shoes	12
Diameter of thrust side rods, inches	4
stern-tube shafting, inches	18½
shaft hole, inches	10½
Length of stern-tube shaft forward bearing, inches	57
after bearing, inches	57½
Diameter of propeller shaft, inches	18½
hole, inches	10½
Length of forward strut bearing, inches	38½
after strut bearing, inches	60½
Diameter of thrust-shaft coupling flange, inches	32
Thickness of thrust-shaft coupling flange, inches	6½
Number of thrust-shaft coupling bolts	12
Diameter of thrust-shaft coupling bolts, inches	3½
pitch circle, inches	26
Length of collar on stern-tube shaft, inches	13
Diameter of collar on stern-tube shaft (inside), inches	18½
Number of keys	3
Length of keys, inches	13
Width of keys, inches	3
Depth of keys, inches	1¾
Thickness of halved ring, inches	2
Inside diameter of halved ring, inches	16½
Length of outboard coupling sleeve, inches	72
Outside diameter of outboard coupling sleeve, inches	24
Width of longitudinal keys, inches	3½
Depth of longitudinal keys, inches	2
Width of cotter keys, top, inches	10
bottom, inches	8 7/16
Thickness of cotter keys, inches	2¼

PROPELLERS.

The hub is fitted on the tapered end of the propeller shaft and held in place by one longitudinal key and a composition nut on the end of the shaft, the nut being threaded the reverse of the direction of the propeller. The nut is locked in place and covered with a composition cap bolted watertight to the hub. The blades fit into tapered recesses in the hub, being held in place by rolled manganese tap bolts, composition chocks being fitted around the bodies of the bolts to keep the blades from shifting after being set to the desired pitch. The bolt holes in the hub are oval in shape and allow a pitch adjustment of one foot either way. The propellers were accurately balanced, and each blade was made a

true surface on both faces, by machining, grinding and polishing. The material for blades and hubs is manganese-bronze.

DATA FOR ONE PROPELLER.

Number of blades	3
Diameter of propeller, feet and inches	18–00
Pitch for official trial, feet and inches	21–09
maximum, feet and inches	22–06
minimum, feet and inches	20–06
Ratio of pitch to diameter	1.2083
Helicoidal area, square feet	98.63
Projected area, square feet	80.72
Disc area, square feet	254.46
Ratio projected to disc area	.317
Area of immersed midship section, square feet	1,732
Ratio disc to immersed midship section area	.1468
projected to immersed midship section area	.046
Number of tap bolts for blades	27
Diameter of tap bolts for blades, inches	3½
Length of tap bolts for blades under head, inches	7
Width of longitudiual key, inches	4
Thickness of longitudinal key, inches	2¼
Length of longitudinal key, inches	39½
Diameter of shaft-nut thread, inches	12
Length of shaft-nut thread, inches	9
Number of threads per inch	4
Immersion of tip of blade, inches	67⅝
Tip of blade above keel, inches	16⅝

DATA FROM OFFICIAL TRIAL.

	Port.	*Starboard.*
Average revolutions per minute on course	124.47	123.3
Slip, per cent. of its own speed	16.65	15.84

OIL AND WATER SERVICE.

Near the after end of each engine room, located on the center-line bulkhead, is a small steam pump, with hand attachment, for supplying oil to the distributing tank and to the oiling system direct if desired. A pipe from this tank connects to all the oil boxes on the main engines and to the oil tanks in the engine room. All of the oil boxes are arranged with sight and wick feeds, and the distributing pipes are jointed in such a manner that they may be taken down easily for cleaning.

For the water service there is in each engine room a 5-inch connection to the main circulating-pump discharge to the

main condenser, with a 4-inch branch to the main engine, and 2-inch to the shaft alley, and smaller branches as follows: Crank-shaft bearings, crank pin, crosshead, crosshead guide, eccentrics, thrust bearing, etc. The discharge from the crosshead guides, main engine and thrust bearings connects to the suction side of the main circulating pump, so as to avoid flooding the bilges.

BOILERS.

The boilers on this ship are of the Babcock & Wilcox type, and were designed, built and installed by this well known concern. The battery consists of sixteen boilers placed two each in eight watertight compartments, all compartments being symmetrically arranged.

Each boiler has one drum extending the full width of the boiler and is located within the casings, so that about one-fourth of the shell is in contact with the gases at the top of the combustion chamber. Immediately below the drum are the front headers, and at the back of the boilers are the back headers, connected by tubes set at an angle of 15 degrees.

The weight of the drum and front headers, etc., is carried by the furnace front, and that of the back headers by a light structural-steel girder, that also forms a portion of the casing.

This is protected from the fire by a bridge wall of brick extending the full width of the furnaces. There are two of these of equal size, separated by a wall of rectangular water legs that extends from the lower row of tubes to a point just below the surface of the grate. The side walls are similarly constructed, and all are connected to a mud drum carried in the boiler front immediately below and connected to the front headers, thereby providing a constant circulation of water and affording excellent means for draining and blowing off.

The boiler casings are made of steel plates and shapes, fire brick and block magnesia being used for insulation. Front and back at the header space they are in the form of doors of such size that they may be easily removed for access to the inside of the tubes, and the sides are removable in sections for access to the baffle plates and for cleaning soot from the tubes.

Each furnace is provided with three furnace and two ash-pit doors, and the grate bars are of the ordinary common type, of such size and shape as to be easily handled by one man.

Boiler Drums.—The boiler drums are of open-hearth steel plate, having two seams which are butt-strapped inside and out. The heads are convex, the radius of which is equal to the inside diameter of the drums, and in each is a flanged manhole shaped from the same plate. All butt straps were formed to the proper curvature of the drum under hydraulic pressure, the rivet holes being punched about ¼-inch smaller than the diameter of the rivets used, and then drilled out full size after the plates were rolled and assembled. After drilling all burrs were cleaned off, the plates reassembled, with turned bolts holding the parts in place for riveting. Wrought-steel pads securely riveted to the drum shell are provided for attaching all valves and fittings.

The following external fittings are provided for each drum:
One 6-inch hand- and steam-closing boiler stop valve.
One twin-spring safety valve, each valve 4½ inches in diameter.
Two 2½-inch stop valves for main and auxiliary feed.
Two 2½-inch check valves for main and auxiliary feed.
Two Klinger reflex water gauges with automatic fittings.
One 1½-inch surface blow valve.
One ½-inch sentinel valve.
One ½-inch air valve.
One ¼-inch steam-gauge connection.
Three ⅜-inch try cocks.

The following internal fittings are provided:
One 6-inch brass dry pipe.
One 2½-inch main feed pipe.
One 2½-inch auxiliary feed pipe.
One 1½-inch pipe and scum pan for surface blow.
Zinc protectors and baskets.

Headers.—The headers are of forged steel, forged in one piece into a rectangular section having male and female sinuous side for nesting when assembled. The front side is provided

with numerous handholes for access to the generating tubes. The covers for these handholes are of pressed steel held in place by one stud, nut and dog. All holes for the generating tubes and connecting nipples between headers and drums are bored and reamed the size of tubing used.

Tubes.—The generating tubes are straight and are made of seamless cold-drawn steel. These and the header and drum nipples, which are of the same material, are rolled in place by the B. & W. patent expander. Only one nipple showed any sign of leak on the trial.

DATA FOR BOILERS.

Number of boilers	16
Length of drum, feet and inches	16–04⅝
Inside diameter of drum, inches	42
Thickness of drum plate, inches	7/16
Heating surface per header and tubes, square feet	164.22
Total heating surface for one boiler, square feet	4,434
grate surface for one boiler, square feet	100
H.S. for all boilers, square feet	70,944
G.S. for all boilers, square feet	1,600
Ratio H.S. to G.S.	44.3–1
Number of boilers per smoke pipe	4
Height of smoke pipe above grate, feet and inches	92–05
Area through smoke pipe, square feet	57.86
Ratio G.S. to smoke pipe area	6.91
Size of generating tubes, inches	2 and 4
Total number of headers, per boiler	27
Number of 2-inch generating tubes per header, 2 with	25
25 with	34
4-inch generating tubes per header	1
Thickness of 2-inch generating tubes, B W.G.	8
4-inch generating tubes, B.W.G.	6
Length of generating tubes, feet and inches	8–00
Number of 2-inch generating tubes per boiler	900
4-inch generating tubes per boiler	27
Inclination of tubes, degrees	15
Height of boiler, feet and inches	13–05½
Length of boiler, feet and inches	9–01½
Width of boiler, feet and inches	16–00½
Total volume occupied by boilers, cubic feet, about	31,000
floor space occupied by boilers, square feet	2,344
area of firerooms in ship, square feet	6,256
volume of firerooms above floors, cubic feet, about	176,000
Ratio of boiler floor space, to fire-room floor area	.374
to fireroom volume.	.176

Weight of one boiler complete, dry, pounds	91,200
water in one boiler, pounds	17,736
one boiler complete, wet, pounds	108,936
Total weight of boilers, dry, pounds	1,459,200
water in boilers, pounds	283,776
boilers, wet, pounds	1,742,976
Weight of boiler, dry, per cubic foot, pounds	47.07
wet, per cubic foot, pounds	56.22
Ratio of weight of boilers dry to boilers wet	.837

OFFICIAL TRIALS.

STANDARDIZATION.

For the purpose of standardizing the screws, runs were made over the measured-mile course off Rockland, Maine, April 11, 1906. The weather was clear and cool with moderate to gentle breezes. Smooth sea.

Previous to the vessel's starting on the standardization runs the draught was taken and found to be—

Forward, feet and inches,	25–01¾
Aft, feet and inches,	24–11¾
Mean, feet and inches,	25–00¾
Corresponding displacement, tons,	14,540

These runs gave the data recorded in Tables I and II, from which the curves on Plate II were plotted.

After the standardization runs the draught was—

Forward, feet and inches,	24–09¾
Aft, feet and inches,	24–09½
Mean, feet and inches,	24–09½
Corresponding displacement, tons,	14,350

FOUR-HOURS' OFFICIAL TRIAL.

The ship left her anchorage at Rockland, Maine, at 10·15 A. M., April 12th, 1906, and steered a straight southeast course out to sea. A period of two hours was consumed in working the machinery up to power, and at 12·15 the signal was given that the four-hour run was begun, which was finished at 4·15 P. M. without the least trouble arising to cause a pause for an instant. The main engines ran with remarkably little vibration, none of the main bearings and journals

Table I.—U. S. S. *WASHINGTON.*

Data from Progressive Runs over Measured-Mile Course off Rockland, April 11, 1906.

Number and direction of run.	Time on course.	Revolutions of main engines.					Speed in knots per hour.	Indicated horsepower.											
		To make 1 knot.		Per minute.				Starboard engine.					Port engine.				Total of both main engines.	Auxiliaries in operation.	Total for all machinery.
		Starboard.	Port.	Starboard.	Port.	Mean.		H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.	H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.			
	Hr. s.																		
1, S.	2–37	322.4	327.1	123.2	125.0	124.1	22.93	3,647.4	5,390 0	2,552.4	2,552.4	14,142.2	3,441.9	5,899.3	2,330.3	2,382.1	14,003.6	28,145.8	787.4	28,933.2
2, N.	2–46.0	342.2	348.1	123.4	125.5	124.5	21.68	3,534.7	5,165.4	2,556.7	2,556 7	13,813.5	3,457.7	5,445 9	2,549.0	2,497.0	13,949.6	27,763.1	825 9	28,589 0
3, S.	2–34.5	322.8	327.1	125.1	126.8	125.9	23.29	3,342.1	5,426.8	2,816.0	2,886.2	14,471.1	3,184.2	5,682 6	2,692.1	2,679.0	14,237.9	28,709.0	855.5	29,564.5
4, N.	2–47.0	343.2	349 1	123.3	125.4	124.4	21.56	3,395.1	5,347.4	2,541.2	2,490.1	13,773.8	3,402.3	5,364.0	2,442.5	2,520.4	13,729.2	27,503.0	847.3	28,350.3
5, S.	2–34.7	322 1	328.3	124.7	127.1	125.9	23 27	3,399.9	5,646.4	2,648.0	2,777.2	14,471.5	3,063.3	5,610 9	2,646 8	2,607.3	13,928.3	28,399 8	851.2	29,251.0
6, N.	2–51.8	338 7	346.3	118.2	120.8	119.5	20.95	4,198.5	4,137.0	2,081.2	2,081.2	12,497.9	4,242.1	4,045 2	1,939 7	1,939.7	12,166.7	24,664.6	701.5	25,366.1
7, S.	2–43.8	316.7	319.0	115 9	116 8	116.4	21.97	3,928.1	3,794.5	1,849.9	1,982.0	11,554 5	3,779 5	3,510.9	1,669.6	1,718.0	10,678.0	22,232.5	669.0	22,901.5
8, N.	3–1.5	336.1	335.3	111.3	111.0	111 2	19.83	3,433 5	3,261.3	1,671.9	1,718.1	10,084.8	3,210.7	2,951.3	1,598.7	1,403.2	9,169.9	19,254.7	617.9	19,872.6
9, S.	2–52 5	306.2	308.9	106.4	107.4	106.9	20.86	2,931.7	2,697.6	1,411.5	1,455.6	8,496.4	2,913.5	2,860 6	1,157.0	1,257.1	8,188.2	16,684.6	578.7	17,263.3
10, N.	3–1.5	333.1	337 2	102.2	103.5	102.8	18.44	2,632.2	2,489 3	1,207.1	1,217.7	7,546.3	2,706.9	2,440.9	1,061.0	1,152.1	7,360 9	14,907.2	463 7	15,370 9
11, S.	3–3.6	303.4	306.4	99.2	100.2	99.7	19.60	2,376 9	2,219.5	1,099.5	1,120.1	6,816 0	2,372.8	2,096.5	913.2	965.1	6,347 6	13,163 6	463.3	13,626.9
12, N.	3–31.9	333.9	338.6	94.5	95.9	95.2	16.99	2,082.5	1,956.6	930.2	979 2	5,948 5	2,125.0	1,896.5	819.1	868.8	5,709.4	11,657.9	408.8	12,066.7
13, S.	3–11.0	316.9	307.1	99.5	96.5	98.0	18.85	2,466.8	2,097.9	1,072.2	1,113.4	6,750.3	1,992.2	1,717.6	679.4	809.3	5,198 5	11,948.8	400.1	12,348.9
14, N.	2–50.7	349.5	331.6	91.1	86.4	88.8	15.60	2,057.4	1,837 3	877.7	877.6	5,650.0	1,499.5	1,348.3	447.6	519.2	3,814.6	9,464 6	378 6	9,843 2

Table II.—U. S. S. *WASHINGTON.*

Data from Progressive Runs over Measured-Mile Course, off Rockland, Me., April 11, 1906.

Number and direction of run.	Which engine.	Steam pressures.			Mean effective pressures.				Vacuum, in inches.	Air pressure, in inches of water.	Cut-off of cylinders from beginning.				Temperatures.					Revolutions or double strokes.							Pyrometers 48 feet above grates.
		In boilers above atmosphere.	At engine above atmosphere.	H.P. steam chest, absolute.	In 1st receiver, absolute.	In 2d receiver, absolute.	H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Referred to L.P.		Outside air.	Engine-room working platform.	Fireroom working level.	Injection.	Discharge.	Hotwell.	Feed.	Main feed pump.	Hotwell pump.	Air pump.	Bilge pump.	Auxiliary condenser pump.	Circulating pump.	Dynamo.	Fireroom blowers.	
1, S.	S.	278	260	230	39.0	107.5	57.5	20.0	20.0	26.5	1.38	H.P.	33	69	63	35	98	54	132	33.0	36	24	40	24.0	150	340	412	595
	P.	288	270	218	37.5	100.0	61.5	18.0	18.4	25.5		I.P.		65	65	35	108	56	146	15.0	28	22		30.0	170			
2, N.	S.	275	265	180	39.0	104.0	55.0	20.0	20.0	26.5	1.57	Full.	33	63	63	35	98	54	134	17.0	38	24	30	30.5	166	340	437	665
	P.	264	265	180	37.5	100.0	57.0	19.6	19.2	26.0		Forward L.P.		68	63	35	104	57	132	17.0	38	23		30.5	166			
3. S.	S.	278	265	180	39.5	97.0	58.0	20.5	20.5	26.5	1.58	Full.	34	69	63	35	107	56	140	21.5	39	23	...	30.5	165	340	442	675
	P.	264	265	183	37.0	91.8	58.0	20.5	20.4	25.5		After L.P.		65	63	35	107	57	140	24.0	39	23	...	30.5	165			
4. N.	S.	277	270	183	39.5	109.0	57.0	19.9	19.5	26.5	1.54		35	73	66	35	107	59	140	26.0	39	25	...	30.5	165	340	436	655
	P.	270	267	181	38.8	98.5	58.8	19.8	19.4	26.0				68	63	35	107	59	142	26.0	39	25		30.5	165			
5. S.	S.	272	265	180	39.0	90.0	59.5	20.3	20.1	26.5	1.55		35	73	66	35	105	57	142	28.0	38	24	40	34.0	170	340	430	665
	P.	267	267	180	39.0	87.5	58.5	20.5	19.8	26.0				68	63	35	107	58	142	28.0	39	23		36.0	175			
6, N.	S.	280	270	270	39.0	109.5	58.0	17.0	17.0	27.0	1.15		36	73	65	35	94	54	138	18.0	38	24	40	38.0	175	340	350	633
	P.	270	260	267	39.0	117.5	55.0	15.5	16.5	26.0				65	65	35	94	54	138	18.0	38	25	40	35.0	174			
7, S.	S.	264	260	259	39.0	117.5	53.5	14.4	14.3	27.0	0.63		36	69	67	35	88	54	138	27.0	38	25	40	36.0	174	340	280	570
	P.	273	260	257	39.0	118.5	53.5	14.5	14.9	27.0				67	68	35	88	54	138	28.0	38	24		36.0	174			
8, N.	S.	273	273	245	39.5	118.5	53.5	14.5	14.9	27.0	0.57		37	71	68	35	87	54	138	29.0	38	24	40	36.0	174	340	261	535
	P.	273	245	230	39.5	105.0	58.5	20.5	20.2	27.0				68	68	35	87	54	138	29.0	38	24		36.0	174			
9. S.	S.	246	245	72	24.0	100.0	33.0	13.8	13.2	27.0	0.33		37	65	67	35	79	45	138	13.5	17	23	20	36.0	164	615	265	530
	P.	245	245	70	23.0	98.5	35.0	10.4	11.3	26.0				68	67	35	87	45	138	9.0	15	24		38.0	164			
10, N.	S.	245	245	205	21.0	93.5	34.0	9.9	10.75	27.0	0.32		38	70	68	35	87	46	140	17.0	18	24	...	30.0	150	615	214	465
	P.	235	190	60	20.0	85.0	31.4	10.7	10.9	26.0				69	69	35	96	46	140	17.0	15	24		36.0	160			
11, S.	S.	238	235	190	20.0	86.0	31.5	8.8	9.3	26.0	0.31		38	73	69	35	78	42	150	14.5	19	24	42	36.0	150	615	216	500
	P.	240	178	55	18.0	80.0	27.5	10.3	9.5	27.0				71	70	35	78	42	150	14.5	19	24		36.0	150			
12, N.	S.	225	230	178	18.0	80.5	27.8	9.25	8.75	26.0	0.0		38	73	70	35	74	42	142	14.5	0	12	20	30.0	152	615	0	475
	P.	248	190	57	19.0	90.5	27.7	10.4	10.8	27.0				73	70	35	74	42	142	15.5	0	24		36.0	144			
13, S.	S.	252	245	180	16.0	75.0	23.4	6.8	8.1	26.0	0.0		39	74	70	35	66	44	163	11.0	0	24	56	30.0	136	615	0	460
	P.	245	175	55	18.0	82.0	25.5	9.3	9.3	27.0				76	70	35	61	41	138	10.0	0	20	28	30.0	136	615	0	458
14, N.	P.	242	175	58	13.0	63.0	20.5	5.0	5.8	26.0	0.0		39	74	70	35	64	44	138	11.5	0	25		36.0	145	615		

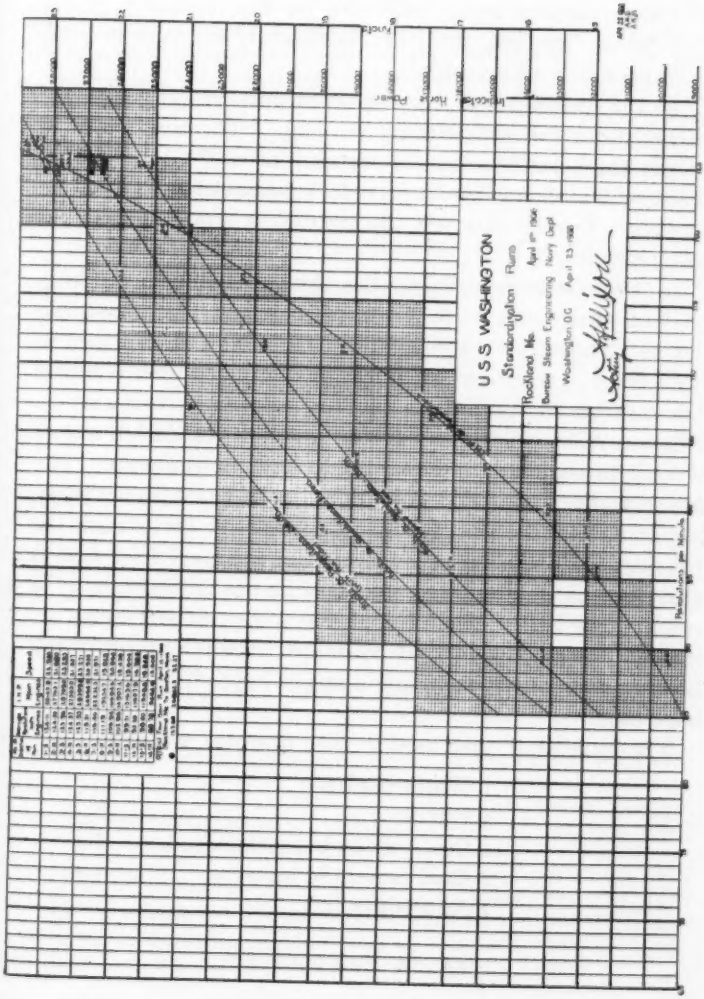

USS WASHINGTON
Standardyton Plate
Rockland Me
April 1st 1908
Builder Steam Engineering Navy Dept
Washington DC April 13 1908

Plate II.

of the gear heating, the smooth running of the valve gear, which seemed to almost float, being particularly noticeable. The water service being used only on the crosshead guides and thrust bearings, furnishes satisfactory evidence of good running and workmanship.

At the conclusion of the four-hour run the engines were slowed down for maneuvering the ship. The time required to put the helm from hard-a-starboard to hard-a-port was ten seconds, and from hard-a-port to hard-a-starboard was ten seconds. During this test the maximum angle of heel was seven degrees.

In the firerooms during the four-hour run everything worked easily and with great regularity. The time-firing device installed by the contractors, the design of two men in their employ, regulated the intervals of firing, which averaged five minutes for each furnace. The performance of this device was most satisfactory and the simplicity of design makes it easy to regulate, overhaul and stow in the ship. The forced-draft blowers and engines ran with noted smoothness, and at no time did the revolutions exceed 410 per minute. In the forward firerooms at intervals the auxiliary feed pumps were used to make up feed, the main pumps not being able to reach those boilers when forced. In order to determine accurately the amount of coal consumed it was put into bags holding 200 pounds each, and a record kept of the number of bags used.

The number of men used in each fireroom was as follows:

1 Water tender.
4 Firemen.
2 Coal passers in fireroom.
2 Coal passers in bunkers.
1 Apprentice boy.

DATA FOR FOUR-HOUR RUN.

Draught at beginning of trial, forward, feet and inches.....	25–04⅝
aft, feet and inches.....	24–11½
Mean draught at beginning of trial, feet and inches.....	25–01 3/16

Corresponding displacement at mean draught at beginning of trial, tons	14,600
Draught at end of trial, forward, feet and inches	25-00
aft, feet and inches	24-09¾
Mean draught at end of trial, feet and inches	24-10¼
Corresponding displacement at mean draught at end of trial, tons	14,457
Corresponding displacement at mean draught during trial, tons	14,520
Speed of ship, in knots per hour	22.27

	Starboard.	*Port.*
Slip of propellers, in per cent. of their speed	15.84	16.65

PERFORMANCE.—FOUR HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Steam pressure at boilers (per gauge), pounds	274.2	
H.P. steam chest (gauge), pounds	248.9	250.4
1st receiver (absolute), pounds	120.0	114.3
2d receiver (absolute), pounds	38.4	36.5
Vacuum in condensers, inches of mercury	25.5	26.0

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	38.8	38.8
Discharge, degrees	101.5	107.0
Hotwell, degrees	59.8	69.0
Feed-water, degrees	135.3	125.0
Engine room, upper platform, degrees	83.7	80.0
working platform, degrees	68.0	82.0
Firerooms, above grating, degrees	68.8	
working level, degrees	68.8	
Smoke pipe, about 48 feet above grate, degrees	697.7	
Atmosphere	40.0	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Main engines (mean for trial)	123.3	124.47
average for both	123.88	
Pumps, main, air	22.5	24.9
circulating	168.0	158.0
feed	24.2	25.0
hotwell	24.8	20.0
bilge	.....	.....
water service	.....	.....
auxiliary feed	.....	.....
condenser	17.2	21.0

	Starboard.	*Port.*
Blower engines	379.8	
Speed of ship, in knots per hour	22.27	
Slip of propeller, in per cent. of its own speed, based on mean pitch and average speed	15.84	16.65
Air pressure in firerooms, inches of water	1.27	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Averages of cards taken in half-hourly periods.*)

Main engines, H.P. cylinder	102.75	103.6
I.P. cylinder	56.75	49.4
F.L.P. cylinder	19.84	17.9
A.L.P. cylinder	20.05	18.7
Mean equivalent pressure, in pounds per square inch referred to combined area of L.P. pistons	54.45	50.25

INDICATED HORSEPOWER.

Main engines, H.P. cylinder	3,487.5	3,551.5
I.P. cylinder	5,324.3	4,677.3
F.L.P. cylinder	2,533.3	2,312.4
A.L.P. cylinder	2,560.5	2,415.5
Total	13,905.6	12,956.7
both engines	26,862.3	

AUXILIARIES.

Air pumps	17.7	19.0
Circulating pumps, main	68.0	66.0
Feed pumps, main and auxiliary	54.0	56.5
Hotwell pumps	5.4	3.4
Bilge pumps	0.0	0.2
Water-service pumps	.....	.....
Auxiliary condenser pumps	1.6	2.1
Blower engines (forced draft)	227.2	
Steering engines (estimated)	.....	
Dynamo engines	80.0	
Total	601.1	
Total all machinery in operation	27,463.4	

COAL.

Kind and quality	Pocahontas, hand-picked, excellent.
Average thickness of fires, inches	8 to 10
Coal burned per hour, pounds	52,010
Average temperature at base of smoke pipe, degrees Fahrenheit	697.7
of atmosphere, degrees Fahrenheit	40.0
air pressure in firerooms, inches of water	1.27

Deduced Data.

Coal burned per square foot of G.S. per hour, pounds	32.5
H.S. per hour, pound	0.733
I.H.P. of main engines per hour, pounds	1.936

Coal burned per total I.H.P. main engines and all auxiliary engines per hour, pounds..... 1.894
Coal burned per total I.H.P. main engines, air, circulating, feed and hotwell pumps, pounds..... 1.915
I.H.P. per square foot of G.S. main engines..... 16.79
and auxiliaries..... 17.16
H.S. per I.H.P. main engines..... 2.64
and auxiliaries 2.58

INDICATOR CARDS.

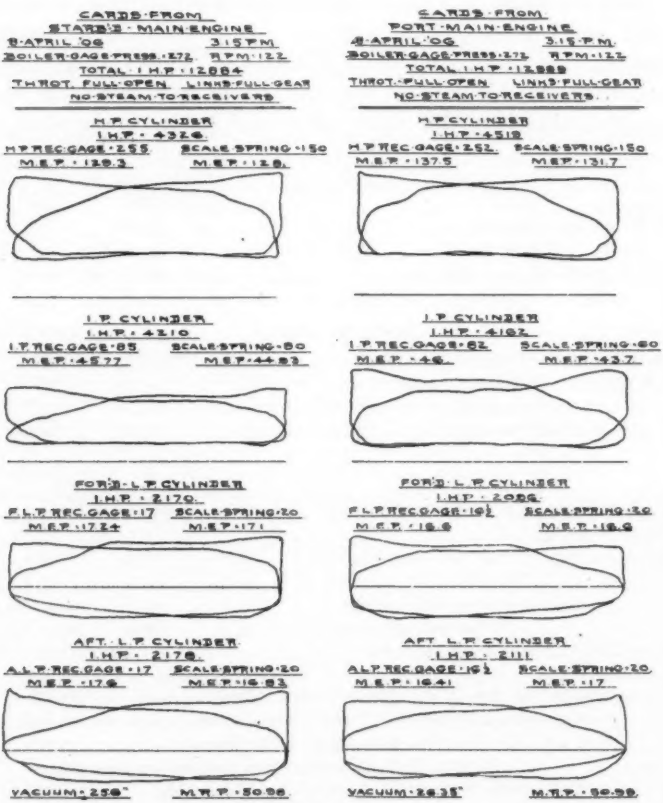
One set of cards representing maximum conditions obtainable under forced draft without steam to receivers were selected for comparison with theoretical cards and average condition on the trial. Accompanying the data will be found a sketch of the cards noted and the same combined.

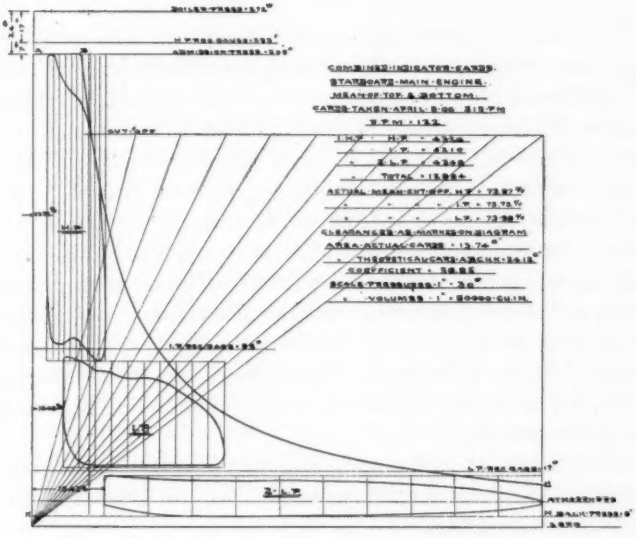
Port Engine.	Cards taken witho't steam to receivers. M.E.P	Cards taken on trial with steam to receivers. M.E.P.
H.P. cylinder, average	134.6	103.6
I.P. cylinder, average	45.85	49.4
F.L.P. cylinder, average	16.6	17.9
A.L.P. cylinder, average	16.7	18.7
Starboard Engine.		
H.P. cylinder, average	128.65	102.7
I.P. cylinder, average	45.35	56.7
F.L.P. cylinder, average	17.17	19.8
A.L.P. cylinder, average	17.21	20.05

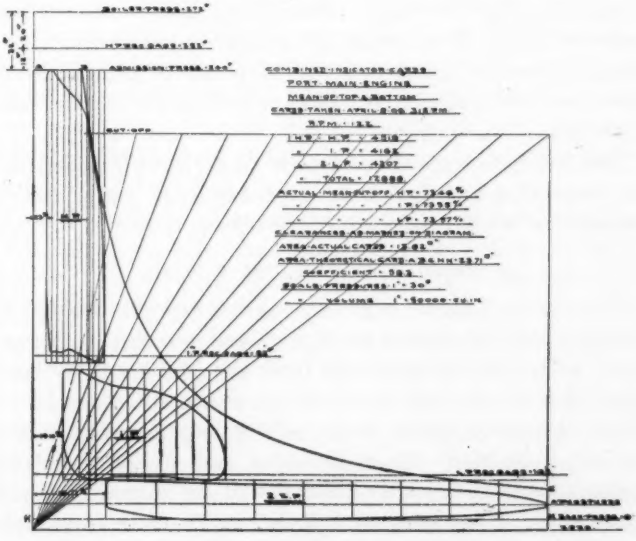
	H.P.	I.P.	Combined F.L.P., A.L.P.
Area of piston, top, square inches	1,164.0	3,167.0	8,602.0
Ratio of areas	1.0	2.72	7.39
Per cent. cut off, top, act. average	77.0	78.25	78.1
bottom, act. average	70.4	69.0	69.4
clearance, top, act. average	23.0	17.8	16.5 14.9
bottom, act. average	24.3	20.5	18.0 16.4

Volume at cut off, top, H.P. cylinder, cubic inches..... 43,021
of clearance, top, H.P. cylinder, cubic inches..... 12,894
at cut off, total H.P. cylinder cubic inches..... 55,915
bottom, H.P. cylinder, cubic inches..... 37,635
of clearance, bottom, H.P. cylinder, cubic inches..... 13,007
at cut off, total, H.P. cylinder, cubic inches..... 50,642
Mean of top and bottom volumes H.P. cylinder, cubic inches..... 53,278
Piston displacement and clearance, top, L.P. cylinders, cubic inches..... 477,630

Piston displacement and clearance, bottom, L.P. cylinders, cubic inches	479,103
Piston displacement and clearance, mean, L.P. cylinders, cubic inches	478,366
Ratio of initial to final volumes of steam	8.97
Areas of actual trial cards	13.78
theoretical cards	23.91
Ratio of actual to theoretical	.576


CARDS FROM
STARB'D MAIN ENGINE
8 APRIL '06 3.15 P.M.
BOILER GAGE PRESS 272 R.P.M. 122
TOTAL I.H.P. 12884
THROT. FULL OPEN LINKS FULL GEAR
NO STEAM TO RECEIVERS
H.P. CYLINDER
I.H.P. 4326
H.P. REC. GAGE 255 SCALE SPRING 150
M.E.P. 139.3 M.E.P. 128.
I.P. CYLINDER
I.H.P. 4210
I.P. REC. GAGE 85 SCALE SPRING 60
M.E.P. 45.77 M.E.P. 44.83
FOR'D L.P. CYLINDER
I.H.P. 2170
F.L.P. REC. GAGE 17 SCALE SPRING 20
M.E.P. 17.24 M.E.P. 17.1
AFT L.P. CYLINDER
I.H.P. 2178
A.L.P. REC. GAGE 17 SCALE SPRING 20
M.E.P. 17.6 M.E.P. 16.83
VACUUM 2.50" M.T.P. 50.98
CARDS FROM
PORT MAIN ENGINE
8 APRIL '06 3.15 P.M.
BOILER GAGE PRESS 272 R.P.M. 122
TOTAL I.H.P. 12888
THROT. FULL OPEN LINKS FULL GEAR
NO STEAM TO RECEIVERS
H.P. CYLINDER
I.H.P. 4518
H.P. REC. GAGE 252 SCALE SPRING 150
M.E.P. 137.5 M.E.P. 131.7
I.P. CYLINDER
I.H.P. 4182
I.P. REC. GAGE 62 SCALE SPRING 60
M.E.P. 46 M.E.P. 43.7
FOR'D L.P. CYLINDER
I.H.P. 2086
F.L.P. REC. GAGE 16½ SCALE SPRING 20
M.E.P. 16.6 M.E.P. 16.6
AFT L.P. CYLINDER
I.H.P. 2111
A.L.P. REC. GAGE 16½ SCALE SPRING 20
M.E.P. 16.41 M.E.P. 17
VACUUM 2.35" M.T.P. 50.98


COMBINED INDICATOR CARDS.
STARBOARD MAIN ENGINES.
MEAN OF TOP & BOTTOM.
CARDS TAKEN APRIL 8'98 3.15 P.M.
R.P.M. 132.
I.H.P. H.P. 4334
" I.P. 4510
" L.P. 4308
" TOTAL 13,994
ACTUAL MEAN CUT-OFF. H.P. 73.87 %
" " " I.P. 73.73 %
" " " L.P. 73.98 %
CLEARANCES AS MARKED ON DIAGRAM
AREA ACTUAL CARDS 13.74"
" THEORETICAL CARD ABCHK 24.18"
COEFFICIENT .5685
SCALE PRESSURES 1" 30"
" VOLUMES 1" 80000 CU.IN.
H.P.
I.P.
L.P.
3 E.P.


COMBINED INDICATOR CARDS
PORT MAIN ENGINES
MEAN OF TOP & BOTTOM
CARDS TAKEN APRIL 8'98 3.15 PM
R.P.M. 132.
I.H.P. H.P. 4318
" I.P. 4196
" L.P. 4307
" TOTAL 12888
ACTUAL MEAN CUT-OFF. H.P. 73.68 %
" " " I.P. 73.33 %
" " " L.P. 73.77 %
CLEARANCES AS MARKED ON DIAGRAM
AREA ACTUAL CARDS 12.82"
AREA THEORETICAL CARD ABCHK 23.71"
COEFFICIENT .583
SCALE PRESSURES 1" 30"
" VOLUMES 1" 80000 CU.IN.
H.P.
I.P.
L.P.
3 E.P.

INSPECTION AFTER TRIAL.

The following parts were examined and found in excellent condition: interiors of all main-engine cylinders; H.P., one I.P. and one L.P. valve and liners on each engine. Balance pistons for above valves. After L.P. and H.P. crank pins of starboard engine. After L.P. crank pin of port engine. Top brass of after L.P. crank-shaft bearing on each engine. Pins and brasses of one crosshead on each engine. Steam and water cylinders, and suction and discharge valves on the main air pumps. Steam and water cylinders of main feed pumps. Steam and water cylinders at random of auxiliary feed pumps. Steam and water cylinders, valves and chambers of hot-well pumps. Steam cylinder and bearings of one blower engine. Main and auxiliary condensers were found to be tight and free from grease.

In all boilers the tubes were found to be straight, the steam drums in excellent condition, as well as the brick and iron work of the furnaces.

The following were the only *items of criticism:* Crank pin of port I.P. slightly scored. Small sand hole in bore of after steam cylinder of starboard air pump. Several small sand holes in liner of water cylinder of pump of port auxiliary condenser. Slight accumulation of mill scale in side boxes of boilers.

The Board expressed themselves as perfectly satisfied with the inspection and had only expressions of praise for the character of work furnished by the contractors.

STEAM AND EXHAUST PIPING.

The auxiliary steam pipes extend fore and aft through the boiler compartments, one on each side of the ship on the outboard side, and are supported from the bulkheads by heavy wrought-steel brackets to which the pipe castings are bolted, proper allowance having been made for expansion by making the bolt holes oval. From the after end of the after boiler compartments to the after bulkhead of the forward compartments these steam lines are 8 inches in diameter, from which

point forward and across the ship, uniting at the center-line bulkhead, it is 6 inches in diameter, thus forming a loop. The boiler connections to this line are 6 inches in all cases, and the arrangement of expansion joints is in a fore-and-aft direction and self contained for each compartment. The following branches are provided for supplying steam to the auxiliaries: Forward—Deck and refrigerating machinery, whistle and siren and heating system; Aft—Dynamos, evaporators and heating system; and in each boiler compartment as follows: Auxiliary feed pumps, blower engines, ash hoist, tube cleaners and sea chests. In the dynamo rooms, on the engine-room forward bulkhead, the line has a nozzle for steam to the dynamos, and aft of these nozzles in the engine rooms is an 8-inch valve having a casting with two nozzles, from one of which is a 6-inch connection to the main steam line in each compartment, while from the other a pipe of the same size leads aft on the outboard and aft sides through the center-line bulkhead thereby making a complete circuit of the machinery space. Stop valves are located at each end of the circuit on the center-line bulkheads and at each athwartship bulkhead.

The distributing castings in the engine rooms have branches for supplying steam to the steering engine, main, auxiliary, fire and bilge and hot-well pumps, main and auxiliary air pumps, circulating, turning and reversing engines, heating system, and for blowing and boiling out main and auxiliary condensers and sea chests.

Main Steam Line.—The main and auxiliary steam lines are the same for the two forward boilers on each side of the ship, but aft of them they are separate and parallel to the after end of the after boiler compartment, the main steam being above the auxiliary and supported from the bulkheads in a similar manner. Steam is supplied from the auxiliary line by one stop valve in each compartment, and the sizes of the line are successively 11, 12 and 13 inches in diameter to the after bulkhead of the after boiler compartments. At this point it runs athwartship to the center-line bulkhead, passing through the dynamo rooms to the main-engine separators located on this bulkhead about midway of the engine rooms, between which

points it is 14 inches in diameter. At the separator is a stop valve that may be operated mechanically from the other engine room. Between the separator and the main-engine throttle valve is a short piece of 13-inch pipe with an expansion joint at the separator end. A 4½-inch nozzle on the separator is provided for steam to I.P. receivers, and a 4-inch branch is led from the cross connection between the main and auxiliary steam pipes to the L.P. receivers.

The boiler stop valves are of the steam-closing type, but may also be closed by hand. For steam closing a piston is provided on the valve stem, steam being taken from the boiler side of the valve. The small valves regulating the supply of steam may be operated from the floor of each compartment or from the adjacent athwartship compartment.

Exhaust Piping.—The main exhaust pipes, 27 inches in diameter, leading fore and aft from the L.P. cylinders, unite in a Y pipe on the main condensers.

The auxiliary exhaust pipes, 9 inches in diameter, parallel the auxiliary steam through the boiler compartments, crossing and uniting at the center-line bulkhead at the forward end of the forward compartments, and in the after boiler compartments, passing through the dynamo rooms, cross the forward end of the engine rooms and unite at the center-line bulkhead, thus forming a loop, stop valves being provided to each end on the bulkhead. It has valves to direct the exhaust steam into either the main or auxiliary condensers, into either L.P. receiver or the feed-water heaters, and in the after boiler compartments to the atmosphere through the escape pipes. At each connection with a condenser there is a stop valve and a spring relief valve opening toward the condenser, and at the escape pipes two stop valves are provided to minimize the chance of air leaks.

PIPES AND FLANGES.

All steam pipes 2 inches and over are seamless-drawn steel tubing. All steam pipes under 2 inches are seamless-drawn copper tubing. All exhaust pipes are of copper, sizes 6 inches and under being seamless-drawn tubing.

Rolled-steel and composition flanges are used, sizes and bolting being in accordance with the Bureau of Steam Engineering Navy standard.

All valves are composition, large pipe fittings on the main steam line being of cast steel, the expansion-joint bodies being of the same material, with glands and bushings of composition.

Copper gaskets were used on the steam, and sheet Gladiator packing on the exhaust lines.

MAIN CONDENSER.

There is one main condenser in each engine room. They are oval in form, the inside dimensions being 9 feet 4 inches high by 5 feet 8 inches wide by 14 feet 0½ inch between tube sheets, the length over the water chests being about 15 feet 3 inches. The shell plating is of steel, the water chests of composition; tubes of composition 70 parts copper, 29 zinc, 1 tin; tube sheets of rolled Muntz metal, and the tube-supporting plates of sheet steel bushed with composition.

The forward water chest, being the one for the entrance and exit of the circulating water, has a division plate on the center line of the condenser. Sizes of inlet and outlet are 21 inches in diameter, the inlet nozzle being on the bottom.

DATA FOR ONE CONDENSER.

Number of tubes	6,292
B.W.G. of tubes	16
Outside diameter of tubes, inch	⅝
Length of tubes between tube sheets, feet and inches	14–00½
Number of supporting plates	2
Cooling surface, measured on outside of tubes, square feet	14,450
Ratio of cooling surface to total heating surface	.397

DATA FROM TRIAL.

	Starbd.	*Port.*
Temperature of injection water, degrees Fahrenheit	38.8	38.8
discharge, water, degrees Fahrenheit	101.5	107.0
Ratio of cooling surface to I.H.P. of main engines	1.04	1.05

AUXILIARY CONDENSER.

There is one Davidson horizontal combined air and circulating pump with surface condenser in each engine room.

DATA.

Diameter of steam cylinder, inches	7
air cylinder, inches	8
water cylinder, inches	12
Length of stroke, inches	12
Diameter of circulating-pump suction, inches	6
discharge, inches	5
air-pump suction, inches	4
discharge, inches	3
Cooling surface of condenser, square feet	600
Number of tubes	600
Outside diameter of tubes, inch	0⅝
Length of tubes, inches	75½
B.W.G. of tubes	16

MAIN AIR PUMPS.

The main air pumps are of twin, vertical-beam type, of Davidson design. They are located on the outboard side of the engine rooms between the main condensers and the feed and filter tank. The discharge is into the filter portion of the feed tank, and there is a 6½-inch cross connection between the suction pipes from the main condensers.

DATA FOR ONE AIR PUMP.

Diameter of steam cylinders, each, inches	14
water cylinders, each, inches	35
Stroke, inches	18
Diameter of steam piston rods, inches	2¾
air piston rod, inches	3½
suction nozzle, inches	13
discharge nozzle, inches	11
Number of suction and discharge valves in each pump	40
Clear area through each valve, square inches	12.42

Valves made of No. 20 B.W.G. rolled manganese-bronze discs, three discs to each valve.

DATA FROM OFFICIAL TRIAL.

	Starbd.	*Port.*
Average strokes per minute	22.5	24.9
vacuum, inches	26.0	25.5
I.H.P.	17.7	19.0

MAIN FEED PUMPS AND FEED SYSTEM.

There are two Davidson vertical, plunger, double-acting, single, outside-packed main feed pumps in each engine room, located on the center-line bulkhead aft of the operating

platform. The suction is from the feed tank connecting pipe, and the discharge from the two pumps after uniting in one common pipe leads forward, passing through a grease extractor, which may be by-passed, and then through the feed-water heater, which also may be by-passed, into the after fireroom, reducing in size as each fireroom is passed.

DATA FOR ONE PUMP.

Diameter of steam cylinder, inches	15
water plunger, inches	10
Stroke, inches	18
Diameter of piston and plunger rods, inches	3
valve, inches	6
suction nozzle, inches	6½
discharge nozzle, inches	6
Material of water end	Composition.

DATA FROM OFFICIAL TRIAL.

	Starboard.	*Port.*
Average double strokes per minute	24.2	25.0
I.H.P. per minute	54.0	56.5
pressure on feed main, pounds	305.0	302.0

FEED-WATER HEATER.

There is one horizontal feed-water heater on the discharge side of the main feed pumps, on the center-line bulkhead at the forward end of each engine room. The material of water heads is cast steel, and that of the tubes, tube sheets and shell, wrought steel.

DATA FOR ONE HEATER.

Total heating surface, measured on the outside of the tubes, square feet	1,366
Total number of tubes	588
Outside diameter of tubes, inch	1
Length of tubes between tube sheets, feet and inches	8–10½
B.W.G. of tubes	16
Thickness of tube sheets, inch	1

DATA FROM OFFICIAL TRIAL.

	Starboard.	*Port.*
Temperature of feed water, degrees Fahrenheit	135.3	125.0
Pressure on feed line, pounds	305.0	302.0
Exhaust steam pressure in heaters, pounds	4.0	1.5

FEED AND FILTER TANKS.

The feed and filter tank is located on the outboard side of the after end of each engine room. It is rectangular in shape, and at the top is arranged as a filter, the filtering material being "loofa," which is placed between perforated plates. The filtering chamber is divided by vertical division plates, arranged so that the water flows under and over in succession. All material is steel, well galvanized before assembling.

Capacity of each feed and filter tank, gallons	6,020
Total capacity of feed and filter tanks, gallons	12,040
Capacity of reserve feed tanks in inner bottom :	
B–92, gallons	3,509
B–93, gallons	3,509
B–94, gallons	4,940
B–95, gallons	4,940
B–96, gallons	4,202
B–97, gallons	4,202
B–98, gallons	4,871
B–99, gallons	4,871
Total, gallons	47,084

HOT-WELL PUMP.

There is one Davidson vertical, piston, double-acting, single, light-service, hot-well pump, located on the center-line bulkhead in the after end of each engine room. Size of pump, 12 inches by 16 inches by 16-inch stroke.

The suctions are from the feed and filter tanks, main condensers, reserve-feed tanks and ship's side, and the discharge is to the feed-pump suction, feed and filter tanks and reserve-feed tanks.

AUXILIARY FEED PUMPS.

In each fireroom, located on the center-line bulkhead, is one Davidson vertical, plunger, double-acting, single, outside-packed auxiliary feed pump. Size of pumps, 11 inches by 8 inches by 12-inch stroke.

The suctions are from the feed-tank connecting pipe, main and auxiliary drains, and in each compartment from the sea, fireroom floors, boiler bottom blows and hose connection on

construction and repair manifold. The discharges are to the auxiliary-feed main, fire main, hose connection and overboard in each compartment.

MAIN CIRCULATING PUMP AND ENGINE.

The main circulating pump and engine is located on the outboard side of the forward end of the engine room. The pump has three sections, viz: the engine-room bilge, main drain and main injection, the latter being 21 inches at the pump and separating into two 15½-inch connections at the ship's side. The valves of the various systems are locked in such a manner that only one may be operated at a time. The machinery specifications required that the pump should discharge 15,000 gallons of water per minute at a speed not exceeding 245 revolutions per minute, and on a test run by the contractors, using a specially-constructed tank and weir at 225 revolutions, 15,512 gallons were discharged, under a 20-foot 7-inch head, at 250 I.H.P.

DATA FOR ONE PUMP AND ENGINE.

Diameter of suction nozzle, inches	21	
discharge nozzle, inches	21	
impeller, inches	56½	
Width of tip of blade, inches	5½	
Diameter of H.P. cylinder, inches	10	
L.P. cylinder, inches	20	
Stroke, inches	10	
Diameter of piston rods, inches	2½	
Length of connecting rods, center to center, inches	25	
Valve travel, inches	3	
Angular advance, H.P., degrees	33	
L.P., degrees	34	
	Top.	*Bottom.*
Steam lead, H.P., inch	1/16	1/8
L.P., inch	5/32	1/16
Cut-off, H.P., per cent. of stroke	0.77	0.715
L.P., per cent. of stroke	0.78	0.71
Diameter of H.P. piston valve, inches	4½	
Size of L.P. slide valve, inches	1½×19½	
Diameter of crosshead pin, inches	3½	
Length of crosshead pin, inches	5	
Diameter of crank pin, inches	4½	

Length of crank pin, inches	6¾
Diameter of crank shaft, inches	4½
Length of bearings, inches	5¾×8 and 8
Crank angle, degrees	90
Area of crosshead slipper, square inches	60

DATA FROM OFFICIAL TRIAL.

	Starboard.	*Port.*
Average revolutions per minute	160	158
I.H.P.	68	66

EVAPORATING AND DISTILLING PLANT.

The evaporator room is located amidships on the berth deck, forward of the engine room hatches, the distillers being in the ventilator trunk to this room on a level with the main deck.

The plant consists of three evaporators and three distillers, having a designed capacity of 23,000 gallons of potable water in twenty-four hours. The distiller circulating, fresh-water and evaporator-feed pumps are located in the evaporator room.

The evaporator shells and heads are of steel plate, the front steam heads for the coils being of cast steel and the back heads of composition. The tubes are brass, rolled into Muntz-metal tube sheets. Each evaporator is provided with steam gauges and relief valves for the shell and coils, also water column, gauge cocks and blow-off valves.

The distillers are vertical and of the Bureau type. The shell is of cast iron, the covers and tube sheets of composition, and tubes of brass, tinned inside and out. The tubes are expanded into tube sheets by rolling, and expansion is taken care of by having the bottom tube sheet made with a flange that works in a stuffing box. The circulating water enters at the bottom, passes through the tubes and is discharged at the top; the vapor enters the shell at a point close to the top tube sheet, the water leaving at a similar point near the bottom tube sheet. Baffle plates are provided for the inlet nozzles for the circulating water and vapor.

The following pumps of the Davidson type are provided: Distiller circulating, size, 8 by 10 by 12 inches, distiller fresh

water, size, 4½ by 4 by 8 inches; evaporator feed, size, 4½ by 5 by 6 inches.

Piping.—Steam for the evaporators and pumps is taken from a reducing valve located in the evaporator room at the terminus of a branch from the auxiliary steam line. The vapor may be sent to the distillers or the auxiliary exhaust line, the evaporators being arranged for operation in single effect. The fresh water from the distillers passes through a meter into a small reservoir tank located in the evaporator room, from which it may drain by gravity to the main and reserve-feed tanks or may be pumped into these tanks or the fresh-water tanks. The distiller circulating pump has an independent connection to the sea, and, in addition to discharging to the distillers, may discharge to the sanitary system or fire main. The evaporator-feed supply is taken from the suction side of the distiller circulating pump.

A test was made of the plant when the ship was lying at anchor in the harbor at Rockland, Me., April 10, 1906. First each evaporator and distiller was given a two-hour trial, and then the entire plant was run for one hour, thereby furnishing an equivalent full-capacity test of three hours. The discharge from the distillers was run into the reservoir tank and a careful record kept of the number of full tanks. The water was tested for purity at frequent intervals with a solution of nitrate of silver, the concentration of water in the evaporators every half hour, and all other conditions recorded every fifteen minutes.

The following are the averages of the data obtained:

Steam pressure in evaporator coils, pounds	46½
shells, pounds	12
at reducing valve, pounds	99
Pump strokes per minute for entire plant:	
Evaporator feed pump	45
Distiller circulating pump	42
Duration of test, hours	3
Output of distilled water, gallons	3,055
Capacity of distilled water in twenty-four hours, gallons	24,440

DATA FOR ONE EVAPORATOR.

Heating surface, tubes only, square feet	392½
Gallons of water evaporated per square foot of heating surface in twenty-four hours	20.76

DATA FOR ONE DISTILLER.

Cooling surface, tubes only, square feet	153
Gallons of water distilled per square foot of cooling surface in twenty-four hours	53.24
Gallons of water distilled in twenty-four hours	8,147
Temperature of sea water, degrees Fahrenheit	35½
fresh water discharge from distillers, degrees Fahrenheit	110
external air, degrees Fahrenheit	35
evaporator room, degrees Fahrenheit	74½
Concentration of water in evaporators by hydrometer test, degrees Fahrenheit	$0\frac{2}{32}$
Quality of water distilled	Good.
Height of water in evaporators above bottom of glass, inch	1
Pump strokes per minute for one evaporator and one distiller:	
Evaporator feed pump	13½
Distiller circulating pump	33½

REFRIGERATING PLANT.

The refrigerating plant is located amidships on the berth deck forward of the forward turret, the refrigerating rooms, six in number, being adjacent to it port and starboard, and the scuttle butts on the gun deck. The rooms are isolated by air locks and are insulated with cork, covered on the inside with zinc, except the floors, which are covered with sheet lead.

The ice machine is of the Allen dense-air type designed for a cooling effect equivalent to three tons of ice in twenty-four hours. The piping for the cooling air is arranged so that it may be sent through the refrigerating rooms and returned to the machine, or through the scuttle butts returning to the machine, or through the ice-making tanks before going to the refrigerating rooms.

During the test of the plant the air was sent through the ice-making tanks, then to the refrigerating rooms and scuttle butts in succession.

DATA FROM TRIAL.

Duration of trial, hours	7¼
Temperature of water to be frozen, degrees Fahrenheit	65
circulating water, degrees Fahrenheit	40
atmosphere, degrees Fahrenheit	68
room, degrees Fahrenheit	75
water in scuttle butts, degrees Fahrenheit	66
drawn from scuttle butts (at end of three hours, degrees Fahrenheit	44
Quantity of ice in twenty-four hours, pounds	1,040
water cooled in twenty-four hours, gallons	1,600
Initial temperature of refrigerating rooms, degrees Fahrenheit	66
Final temperature of Admiral's room, degrees Fahrenheit	24
Captain's room, degrees Fahrenheit	21
ward officers' room, degrees Fahrenheit	25
junior officers' room, degrees Fahrenheit	16
warrant officers' room, degrees Fahrenheit	14
crew's rooms, degrees Fahrenheit	30
Total wall area of rooms, square feet	1,790
cooling surface of coils, square feet	775
Steam pressure, pounds	107
Compressor-cylinder pressure, pounds	253
Expander-cylinder pressure, pounds	71½
Revolutions per minute	101

MACHINE DATA.

Diameter of steam cylinder, inches	10
compressor cylinder, inches	9
expander cylinder, inches	7½
piston rods, inches	2
Length of connecting rod, center to center, inches	35
Stroke, inches	14

ELECTRIC PLANT.

The electric plant is located below the berth deck in two compartments, port and starboard, of the center ammunition passage at the forward end to the engine rooms. The General Electric Company's marine generating sets are installed, there being six machines, all told, of 100-kw. capacity each, three being in each room.

Steam is taken from the auxiliary steam line on each side of the ship, with a cross pipe connecting the two rooms, and in each room is a combined air and circulating pump and condenser for the exclusive use of the dynamos. There is a

steam separator in each room which drains to a trap and to the steam end of the air pump. The exhaust is into either the dynamo condenser, main or auxiliary condensers or atmosphere.

DATA.

Type of engines	Vertical cross-compound.
Number of revolutions per minute	350
Steam pressure, pounds	150
Diameter of H.P. cylinder, inches	10
L.P. cylinder, inches	18
Length of stroke, inches	10
Diameter of piston rods, inches	2¼
Number of generators	6
Capacity of each generator, in kilowatts	100
Dynamos	10 pole, 125 volts.

DATA FOR ONE DYNAMO CONDENSER.

Make	Davidson.
Type, Horizontal, combined air and circulating pump, with surface condenser.	
Diameter of steam cylinder, inches	8
air cylinder, inches	9
water cylinder, inches	14
Length of stroke, inches	12
Cooling surface of condenser, square feet	700
Number of tubes	701
Outside diameter of tubes, inch	¾
Length of tubes, inches	75½
B.W.G. of tubes	16

ASH HOISTS.

The ash-hoist engines were designed and built by the contractors. They are located entirely within the ventilator trunks, and are designed to stow in one corner of the trunk within a plane not exceeding twelve by eighteen inches. The hoist will deliver to and may be operated from either the main or upper decks, the bucket-guide rails extending from the bottom of the ventilator trunk to a point well above the upper deck, with trolleys at the main deck for delivering ashes to the chutes at the ship's side. On the test 300 pounds was hoisted from the fireroom floor to the upper deck in five seconds, including starting and stopping, with 100 pounds steam pressure.

The principle of the engine is a long cylinder taking steam

at both ends through a piston valve that is controlled by the operator through the medium of a spiral spindle worked by a wheel and gearing outside the ventilator trunk. The spiral spindle works in a nut carried by the crosshead, thus putting the valve automatically on the center and stopping the engine when the operator stops turning. At the proper height above the steam cylinder is the top sheave bracket, between which, on guide rods, the crosshead travels. This crosshead is fastened to the piston rod and carries multiple sheaves corresponding to those carried by the top bracket that give a maximum bucket travel of 52 feet for a piston travel of 6 feet 6 inches.

DATA FOR ONE ENGINE.

Diameter of cylinder, inches	7
Stroke, inches	78
Diameter of piston valve, inches	1¾
Travel of piston valve, inches	2
Diameter of piston rods, inches	1⅝
rope sheaves, inches	8
Pitch of spiral spindle, inches	13
Number of rope sheaves	3
ash-hoist engines in ship	8

FORCED-DRAFT BLOWERS.

The forced-draft blowers are located in the firerooms, being suspended from the protective deck about over the center of the stokehold. The engines are designed and built by W. D. Forbes & Co. and the fans by the contractors.

The arrangement is the same for all of the firerooms, there being two engines, each driving a fan located athwartship, so that the air is discharged directly to the stokehold floor at a distance at about two feet from the front of the boilers. The supply is taken from the fireroom ventilator ducts, which are closed at the bottom when under forced draft.

DATA FOR ONE BLOWER ENGINE.

Number of cylinders	2
Diameter of H.P. cylinder, inches	5
L.P. cylinder, inches	10

Stroke, inches	6
Diameter of H.P. valve, inches	2
L.P. valve, inches	4
Valve travel, inches	1½
Diameter of piston rods, inches	1¾
valve stems, inch	0¾
crank shaft, inches	3
pins, inches	3
fan shaft, inches	2¾
Length of connecting rod between centers, inches	15
Number of bearings	4
Total length of bearings, inches	22½
Crank angle, degrees	180
Base area of engines assembled, inches	30×30
Height of engines assembled	54½

DATA FOR ONE BLOWER FAN.

Diameter of fan, inches	66
Number of blades	10
Width of tip of blades, inches	18½
Area of induction nozzle, square inches	1,303
eduction nozzle, square inches	2,794

DATA FROM OFFICIAL TRIAL.

Average revolutions per minute	380.0
air pressure in firerooms	1.27
I.H.P. for one blower	14.2
sixteen blowers	227.2

THE HEATING SYSTEM.

The heating system is divided into two sections and fourteen circuits, eight circuits being on the forward section and six on the after section. Six of the forward and five of the after circuits are from general manifolds, the other three, being circuits that are constantly under steam, are connected directly to the auxiliary steam line. Steam is supplied to each section through a reducing valve and manifold which admits steam to the several circuits, the drainage being back through a second manifold to traps which discharge to either the feed tanks or main and auxiliary condensers at will. Each circuit has a a stop valve at the steam and drain end so that it may be entirely isolated and water from the other circuits prevented from backing up in it. All pipes are of seamless-drawn brass

tubing with the valves and fittings of composition, and the radiators are made up in sections so that steam may be only on a portion of it at one time if desired.

MACHINE SHOP.

The machine shop is located amidships on the berth deck between the after mast and the continuation of the after engine-room bulkhead, access to it being through the engine-room hatches. The machines are arranged with line and counter shafting driven by a 15-horsepower motor. A test was made to determine whether the motor was sufficiently powerful to do the work, and, with all machines working on hard cast iron at a heavy cut, 5.85 horsepower was developed.

The following machines are installed:

1 28-inch extension-gap lathe.
1 14-inch engine lathe.
1 16-inch crank shaper.
1 31-inch radial drill press.
1 16-inch sensitive drill press.
1 30-inch grindstone.
1 universal milling machine.
1 12-inch emery grinder.
1 combined hand punch and shears.
6 bench vises.

The tools are provided with the most modern attachments, including scroll and drill chucks, index head, automatic cross feed, swivel table, pipe vises, etc., and all necessary tools, drills and cutters.

U. S. BATTLESHIP *GEORGIA.*

DESCRIPTION AND TRIALS.

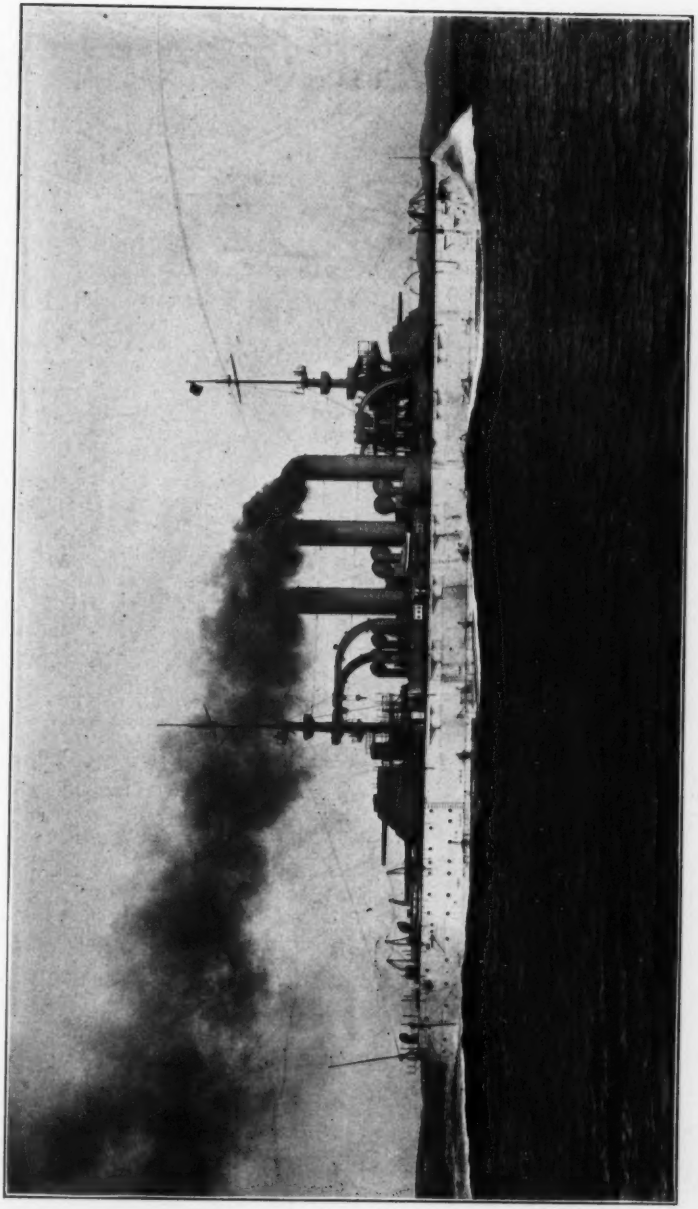
By Clifton Lee, Jr., Draughtsman, U. S. N., Associate.

The U. S. Battleship No. 15, the *Georgia*, is one of three battleships authorized by Act of Congress approved March 3, 1899, the contract being signed by the Bath Iron Works, Bath, Maine, February 18, 1901, for delivery to the Government February 18, 1904. Later three extensions were granted, until March 12, 1905, March 12, 1906, September 18, 1906, respectively.

The contract requires a speed of at least 19 knots per hour, to be maintained for four consecutive hours, on a mean draught of 23 feet 9 inches, corresponding to a displacement of 14,680 tons; and a total cost of $3,590,000, exclusive of armament and certain portions of armor, of which $1,240,000 is for machinery. The allotment of weights for machinery was 1,563 tons, water 167 tons, and machinery with water 1,730 tons. The keel was laid August 31, 1901, and the vessel launched October 11, 1904. A novel feature of the launching of this vessel was that the smoke pipes were in position and steam on some of the boilers, and her whistles were blowing as she went off the ways.

HULL.

The hull is constructed entirely of basic open-hearth steel, with 4-foot frames, spacing and intermediate frames under machinery space. The inner bottom extends between frames 13 and 95, the space between frames 60 and 68½ being used for reserve fresh water. A coffer dam, 36 inches wide, extends the entire length of the ship on protective deck and projects 3 feet above berth deck, that portion forward and abaft the midship armor being packed with cornpith cellulose.



U. S. BATTLESHIP "GEORGIA."

HULL DATA.

Length between perpendiculars, feet	435
over all, feet and inches	441-01½
on load-water line, feet	435
Breadth, extreme, feet and inches	76-01½
molded, feet	76
Depth, feet and inches	41-03¾
Draught, normal, feet and inches	23-09
load, feet and inches	26-03
Ratio of length to beam	5.71
Displacement, tons	14,970
load, tons	16,750
per inch normal draught, tons	60.8
Area of midship section, square feet	1,730
load-water line plane, square feet	25,500
of wetted surface, square feet	42,450
Center of gravity L.W.L. plane aft of M.S. (54½ feet)	2.44
buoyancy above bottom of keel, feet	13.00
forward of midship section, feet	.64
Transverse metacenter above center of buoyancy, feet	19.10
Coefficient of fineness, block	.68
M.S.	.96
L.W.L.	.77
cylindrical	.71
Number of frames (spaced 4 feet)	108
watertight compartments	312

Upper Deck.—This deck, forming the upper part of superstructure, extends fore and aft between the 12-inch turrets about 50 feet wide, leaving a spacious gangway on either side of main deck, and it is cut away to clear the 8-inch broadside turrets. This deck carries the forward bridge, after bridge and eight of the boats.

Main Deck.—The main deck extends the full length of vessel and is surmounted by superstructure as described above. The two 12-inch turrets, with the superposed 8-inch turrets, and two 8-inch broadside turrets are on this deck, and in the superstructure are four 3-inch rapid-fire guns. The hawse pipes enter this deck and the anchor chains are led over deck to windlass house, which projects through deck. A bill board and crane for each of the sheet anchors, Navy type, are on this deck. The bower anchors are of the Baldt type. On this deck will also be found the crew's galley, bakery, officers'

galley, paymaster's office, executive officer's office, officers' water closets, and wireless telegraph station.

Gun Deck.—The 6-inch battery of twelve guns, eight 3-inch R.F. guns, four forward and four aft, are on this deck. Forward are the crew's shower baths and lavatories, amidship the berthing space, and aft the quarters of the admiral, captain, chief of staff, executive officer, navigator, chief engineer and paymaster.

Berth Deck.—On the berth deck will be found, beginning forward, sick bay, operating room, isolation ward, dispensary and bath, machinists' and petty officers' quarters and baths, crew's space, refrigerating room, laundry, firemens' showers and wash rooms, distilling plant; and aft, the warrant, junior and wardroom officers' quarters and baths.

Protective Deck.—This deck extends full length of the ship, and is 3 inches thick on the slope and 1½ inches on the flat. The space between protective and berth decks and coffer dam previously described is devoted to stores, except abreast the firerooms, where it is used for coal.

Lower Decks.—On the upper platform are magazines, handling rooms, storerooms, upper dynamo rooms, engineer's workshop and steering-engine room. On lower platform are magazines, handling rooms and storerooms. In hold are storerooms, magazines, lower dynamo room, coal bunkers, engine and boiler compartments and trimming tanks.

Coal Bunkers.—There are twenty-four coal bunkers, eighteen in the hold and six on the protective deck, with a total capacity of 1,909 tons at 43.5 cubic feet to the ton.

VENTILATION.

The ventilation for quarters outside of armor, crew's space, water closets, etc., is of the exhaust system, and that for quarters, inside armor, engine rooms, dynamo rooms, magazines, etc., is of the supply system. There are thirty-three independent systems, each supplied by an electrically-operated fan, thereby avoiding the piercing of important watertight bulkheads with ventilation pipes. Automatic ball valves are

fitted in all ventilation pipes where they pierce decks or flats below the protective deck.

DRAINAGE AND FLOODING SYSTEMS.

The main drain of 15½-inch diameter steel tubing, galvanized, extends fore and aft on starboard side of center-line bulkhead through firerooms, with crossover connection and valve to opposite firerooms. Just forward of the main engines the main drain separates into two athwartship branches, one to each side of ship, and these branches lead aft, just outboard of main-engine foundations, to the main circulating pumps. All main drain valves, except those at circulating pumps, are operated from deck plates at berth deck and also by handwheels fitted to valve stems 30 inches above floors.

The auxiliary drain of 5½-inch diameter steel tubing, galvanized, extends fore and aft on port side of center-line bulkhead throughout length of machinery space, and is provided with an independent bilge connection to drainage manifold, and a by-pass to main drain in each compartment in machinery space. The drainage manifolds in firerooms are connected to auxiliary feed pump and those in engine rooms are connected to fire and bilge pumps, making a total of ten direct-acting steam pumps and two centrifugal steam pumps that may be used for pumping and drainage. Double-bottom compartments under machinery, except the four used as reserve-feed tanks, are provided with both suction and flooding connections; other double-bottom compartments with suctions only.

The fire mains, of 6-inch copper pipe, sabined, are led fore and aft through machinery space, one on each side of ship, with cross connections at each end, and single mains leading forward and aft from these cross connections to risers and plugs throughout the ship. A connection is provided to the magazine flooding system, and a 5-inch by-pass to flushing system.

The fire main is supplied by risers from the two engine-room fire and bilge pumps and the eight fireroom auxiliary feed pumps.

The flushing system is of 5-inch copper pipe, sabined, and furnishes salt water throughout the ship. It is supplied from the distiller circulating-pump discharge beyond the distillers. The fresh-water service throughout the ship is provided by means of gravity tanks on gun and upper deck, of 13,500 gallons capacity. Two electrically-operated reciprocating pumps are provided, one forward and one aft, for general service of these tanks.

STEERING ENGINE AND GEAR.

The steering engine is of Hyde Windlass Company's design, with two steam cylinders 16 inches in diameter and a 12-inch stroke, taking steam at 100 pounds per square inch, and operating the steering gear through right-and-left-hand screws and traveler with side rods to crosshead. The steering engine is operated from flying bridge, conning tower and steering-engine room through wire-rope transmission. A hand-steering gear is provided in steering-engine room, consisting of three hand wheels driving the screw shaft through a train of gears.

ANCHOR WINDLASS.

There is one Hyde Windlass Company's steam windlass with four wildcats, capable of hoisting two 16,500-pound anchors at once at a speed of six fathoms per minute. The windlass is driven by a double upright engine, with steam cylinders 16 inches in diameter and 14-inch stroke, located on gun deck and connected to windlass through vertical shaft and worm-and-bevel gears. The engine takes steam at 100 pounds pressure.

INTERIOR COMMUNICATION.

A complete system of interior communication is installed by Chas. Cory & Son, of New York, consisting of—

Officers' calls, with bells, buzzers and annunciators;

Telephone system, with 40-call central station in dynamo room;

Voice tubes (135), with electric calls at each end, those beneath protective deck having watertight mouth pieces;

Fire alarm system, with thermostats in 108 compartments, indicating at captain's orderly's station, those in storerooms indicating at 200 degrees, and those in magazines at 110 degrees;

General alarm system, consisting of thirty 12-inch gongs operated from four stations;

Watertight-door alarms, consisting of twenty-three solenoid whistles, operated from two stations;

Electric revolution indicators from shaft alley to bridge and conning tower, indicating the direction and speed of each engine;

Electric and mechanical telegraphs from engine rooms to bridge and conning tower;

Mechanical telegraphs from engine rooms to firerooms;

Electric steering telegraph from bridge and conning tower to steering-engine room;

Electric helm indicator from steering-engine room to conning tower and bridge;

Electric and mechanical log, the electric log registering in chart house, and the mechanical on quarter deck;

Information system from range finders on military masts to battery-control station;

Battery-control system consisting of 1,600 5-candlepower lights, in six groups, indicating all guns and ammunition hoists and giving range and deflection of guns, operated from conning tower and central station in dynamo room.

ARMOR.

The side belt is a complete water-line belt 8 feet in width. This belt is of maximum thickness for a distance of 192 feet abreast the engines and boilers, 11 inches thick at its upper edge, and maintaining this thickness downward for a distance of 5 feet, from which point it tapers to 8 inches thick at its lower edge. This belt is completed to the bow and stern as follows: for a distance of 60 feet forward and 32 feet abaft the heavy belt above described the plates are 9 inches in thickness at the upper edge, tapering uniformly to 6 inches in thick-

ness at the lower edge. The next course of plates forward and aft for a distance of 16½ feet, are 6 inches in thickness at the upper edge, tapering uniformly to 4½ inches at the lower edge. The next course of plates forward and aft for a distance of 17 feet, are 5 inches at the upper edge, tapering to 4 inches at the lower edge. The belt is completed to the bow and stern with plates of a uniform thickness of 4 inches.

Above the main belt, and covering the central portion of the hull for a distance of 245 feet, the ship's sides are protected by armor of a uniform thickness of 6 inches, extending up to the main deck, this upper and lower casemate side armor being joined to the barbette of the after 12-inch turret by athwartship armor 6 inches thick, and connected forward by inclined armor 6 inches thick, thus forming a central casemate. The armor on conning tower is 9 inches in thickness, and that on signal tower is 5 inches.

ARMAMENT.

4	12-inch B.L.R., 40-caliber;	Range of fire, 270 degrees.
8	8-inch B.L.R., 45-caliber;	Range of fire, 270 degrees.
12	6-inch R.F., 50-caliber;	Range of fire, 120 degrees.
12	3-inch R.F., 50-caliber;	Range of fire, varying.
12	3-pdr. semi-automatic guns.	
5	.30-caliber Colt's automatic rifles.	
2	.30-caliber Gatling's machine guns.	
4	21-inch submerged torpedo tubes.	

COMPLEMENT.

Admiral,	1	
Chief of staff,	1	
Commanding officer,	1	
Wardroom officers,	19	
Junior officers,	20	
Warrant officers,	8	
Total,	—	50

Seaman branch,	343	
Artificer branch,	38	
Engine-room branch,	240	
Special branch,	13	
Commissary branch,	15	
Messmen branch,	26	
Aid for flag,	37	
Marine branch,	60	
Total,	—	772
Total, officers and men,		822

PROPELLING MACHINERY.

There are two vertical, inverted, four-cylinder, triple-expansion engines of 19,000 estimated I.H.P. placed athwartship of each other in separate watertight compartments. In the construction of these engines the general design of the Department was carried out, and when engines were nearly ready for erection the contractors were authorized to change them from inboard- to outboard-turning engines. This was easily accomplished by shifting the crosshead guides and strongbacks from outboard to inboard side of engine and a few minor changes in valve gear.

The order of the cylinders is F.L.P., H.P., I.P., A.L.P., the sequence of cranks being H.P., I.P., F.L.P., A.L.P. The cylinders of each engine are supported on twelve forged-steel columns firmly braced by symmetrically-arranged horizontal and diagonal tie rods. The H.P. and I.P. cylinders are connected by guide straps which prevent athwartship motion while permitting fore-and-aft expansion, the other cylinders being rigidly bolted together. The crankshafts are made in two sections, the forward section having the F.L.P. and H.P. cranks at an angle of 180 degrees from each other, the after sections having the I.P. and A.L.P. cranks at an angle of 180 degrees, the two sections being bolted together to give the sequence above named. The H.P. cylinder is provided with one piston valve, the I.P. with two piston valves and the

L.P's. each with one double-ported slide valve. All valves are fitted with balance pistons, those of the H.P. and I.P. valves working in cylinders cast in upper valve-chest cover, and those for the L.P. valves in separate cylinders bolted to top of valve chests.

Engine Training.—In erecting these engines the second and fifth pairs of columns were erected perpendicular to the bedplate, and strongback and tie rods rigidly bolted to them. The other four pairs have the strongback attached to them with a flexible connection to allow a fore-and-aft movement to columns when engines are warming up. These columns, the cylinders, crank-pin brasses, and the valve gears, are erected out of line, in a direction opposite to the direction of expansion an amount equal to about one-half the calculated amount of expansion they will be required to take. By this method of erecting, the engines are in practically the same alignment when under steam as when cold.

MAIN ENGINE DATA.

	H.P.	I.P.	L.P.
Main steam pipe, seamless-drawn steel, diameter, inches			12
Cylinders, number for each engine	1	1	2
diameter, inches	35	57	66
stroke, inches	48	48	48
valves, No. of	1	2	2
kind of	piston.	piston.	slide.
diameter, inches	20	22	...
valve stems, diameter, inches	3	3	3½
piston rods, diameter, inches	8	8	8
diameter axial hole, inches	1½	3½	5
length, feet and inches	...	9-4½	...
cylinder walls (2), thickness of, inches	¾	1½	1¼
liners, thickness of, inches	1½	1⅝	1⅝
jacket space, inch	⅝	⅝	⅝
valve-chest liners, thickness of, inches	1¼	1¼	1⅛
Crosshead surface, ahead, square inches		690	
backing, square inches		510	
Connecting rod, between centers, feet		8	
diameter, inches		Taper 7½ to 9	
diameter axial hole, inches	1½	4½	5
Clearance of all pistons (designed), top, inch		⅛	
bottom, inch		¼	

Crosshead pin, diameter, inches	10
length, inches	10¾
Crank shaft, number of sections	2
diameter at eccentric seat, inches	18
of axial hole, inches	10¼
coupling disc, inches	32
thickness of coupling disc, inches	3¾
number of bolts in coupling disc	8
diameter bolts in coupling disc, at center, inches	3½
diameter of journals, inches	17½
length of journals forward to aft, ins.	25¾, 22, 22, 25, 29½, 25¾
forward section, feet and ins.	16–02½
after section, feet and inches.	17–07
pin, length of, inches	21½
diameter of, inches	18
axial hole, inches	10¼
webs, width of, inches	20
thickness of, inches	11½

Valve Gear.—The valve gear is of the Stevenson type with double-bar links, all valves being worked direct, the two I.P. valves being actuated through a crosshead. The reverse arms are graduated to show the position of cut off from .5 to .73 of stroke.

Reversing Engines.—The engines are reversed by means of a 15-inch diameter by 21½-inch stroke steam cylinder, with an 8½-inch oil-control cylinder, directly connected to an arm on reverse shaft.

Turning Engines.—The after crank-shaft coupling is fitted with a worm wheel driven through worm gear by a 6-inch by 6-inch double, upright engine, a speed of 400 R.P.M. on turning engine giving a speed of .277 R.P.M. on main engines. A ratchet is provided for hand turning.

Shafting and Bearings.—The shafting, class A machinery forgings, is divided into four sections, viz: thrust, line, stern-tube and propeller shafts. The usual sectional ring and shoulder flange is fitted to forward end of stern-tube shaft to facilitate with drawing through stern tube. The outboard coupling is the usual sleeve coupling with taper fit and cross keys. The shafting is encased with composition sleeves in

way of bearings only, and galvanized-steel fairwater, filled with rosin and pitch, are fitted at stern tube and on sleeve coupling.

Thrust shaft, diameter, inches	16¼
axial holes, diameter, inches	10¼
number of collars on each	13
collars, diameter, inches	26
thickness, inches	02
space between, inches	04
length, feet	15
Line shaft, diameter, inches	16¼
axial holes, inches	10¼
length, feet	29
Stern-tube shaft, diameter, inches	17¼
axial hole, inches	10¼
length, feet and inches	36–10
Propeller shaft, diameter, inches	17¼
axial hole, inches	10¼
length, feet and inches	25–06

Thrust Bearing.—A horseshoe thrust bearing with thirteen shoes babbited on both sides, except forward one, which takes ahead thrust only, and cored for circulating water. Screwed side rods, 3¾ inches diameter, and lock nuts are fitted for individual adjustment of horseshoes, and lugs and wedges are fitted for fore-and-aft adjustments of block on sole plate. A steady bearing is provided at each end of block, and interior is used as an oil well and is fitted with cooling coil. Brass covers are fitted between the horseshoes to prevent throwing of oil.

Thrust-bearing surface, ahead, square inches	2,445.43
astern, square inches	2,257.32

Spring Bearings.—There are two spring bearings to each line shaft, 26 inches long, of cast iron and lined with white metal.

Stern-Tube and Strut Bearing.—Each shaft is provided with an inner and outer stern-tube bearing and one strut bearing, consisting of composition sleeves made in halves with lignum-vitæ bearings.

Propellers.—The propellers are three-bladed adjustable pitch,

from 18 feet to 20 feet, of manganese-bronze, and turn outboard when going ahead. They are designed as true screws, and the surfaces ground true and tinned all over.

Diameter of propeller, feet	18.00
Pitch as set, mean, feet	19.17
Helicoidal area, feet	97.92
Disc area, feet	254.47
Pitch ratio, $\frac{D}{P}$	.9389

Main Air Pumps.—In the forward end of each engine room there is one Blake twin, vertical-beam, single-acting, air pump with phosphor-bronze disc valves. This pump is located below and just inboard the main feed tanks and close to main condenser, giving short leads for both suction and discharge pipes. The suction is provided with a 1½-inch connection to the hotwell-pump suction from reserve feed tanks for making up feed.

Steam cylinders (two), diameter, inches	14
Water cylinders (two), diameter, inches	32
Piston rod, steam, diameter, inches	02¾
water, diameter, inches	03½
Stroke, inches	21

Main Condensers.—There are two cylindrical condensers, one in each engine room, with steel shell, composition water heads, and tubes of brass. They are of the return-tube type with grid valve in water head for by-passing water directly overboard.

Diameter of shell, inside, feet and inches	6-10
Thickness of shell, inch	00⅝
Length over heads, feet and inches	15-04
Tubes, number in each condenser	5,583
diameter, inch	00⅝
thickness, B.W.G	18
Length between tube sheets, feet	13
Spacing between centers, inch	00 1⅛
Sheets, thickness, inch	01
Cooling surface, each condenser, square feet	11,875

Main Circulating Pump.—In each engine room just aft the main condenser there is a double-inlet centrifugal circulating pump driven by a vertical, inverted, compound engine with cranks at right angles and counter-balanced. These pumps are designed to discharge 12,000 gallons of water per minute from the bilge at 266 revolutions per minute. These pumps draw from the sea, the main drain and the engine-room bilge, the valves are fitted to open and close only with aid of a key, and it is necessary to close valve before removing key. With only one key it is impossible to have more than one valve open at a time. They discharge through main condensers overboard, or overboard direct through grid valve in condenser water head. The engines take steam from the auxiliary steam pipe and exhaust into the auxiliary exhaust line. A connection is provided for taking the exhaust from L.P. jackets of the main engines to the L.P. receiver of those engines.

Steam cylinder, H.P., diameter, inches	10
L.P., diameter, inches	16
Stroke, inches	10
Pump runner, diameter, inches	42
width at periphery, inches	06½
at hub, inches	10⅜
number of rams	06
Suction, diameter, inches	20
Discharge, diameter, inches	20

Auxiliary Condensers.—On the after engine-room bulkheads, above the grating, is a Wheeler Admiralty Surface Condenser with a Blake simplex combined air and circulating pump attached. They have cylindrical steel shells and composition heads, tube sheets, glands and ferrules. The tubes are of composition, tinned inside and outside.

Condenser, diameter shell, outside, inches	23½
length between tube sheets, feet and inches	8–06
tubes, length, feet and inches	8–08
diameter, outside, inch	00⅝
thickness, B.W.G	18
number of	432
cooling surface, square feet	600

Steam cylinder, diameter, inches	08
Air cylinder, diameter, inches	10
Circulating cylinder, diameter, inches	10
Stroke, inches	12

The connections to these condensers from auxiliary exhaust are provided with a stop valve and a spring-loaded relief valve for use when running the auxiliary exhaust line under pressure.

Hotwell Pump.—In the forward outboard corner of each engine room there is a hotwell pump of the Blake vertical, simplex, piston pattern so piped that it may draw from main feed tanks, reserve feed tanks, main condenser, and from side of ship for taking aboard fresh water, and discharge to main feed tanks, reserve feed tanks and feed suction main.

Steam cylinder, diameter, inches	12
Water cylinder, diameter, inches	14
Stroke, inches	12

Fire and Bilge Pumps.—In the after end of each engine room there is a fire and bilge pump of the Blake vertical, simplex, piston pattern. These pumps are provided with suctions from the sea, the main drain, the auxiliary drain, the drainage manifolds, the engine-room bilge and the crank pits. They discharge to the fire main and overboard.

Steam cylinder, diameter, inches	12
Water cylinder, diameter, inches	10
Stroke, inches	12

Feed and Filter Tank.—In the forward outboard corner of each engine-room, close under protective deck, is a combined feed and filter tank of 5,000 gallons capacity. The upper portion is fitted with three filtering chambers, filled with loofa, so arranged that the pump discharges pass downward through the three filtering chambers successively and overflow to feed tank proper. The suction connection from feed tank is provided with balanced flap valves actuated by a ball float to prevent pump suction from filling with air when water is low in tank. A vapor pipe connects the filter chambers to

the escape pipe at after stack. The feed pumps draw from an equalizing pipe between the two tanks.

Feed Pumps and Piping.—There are sixteen feed pumps of the Blake vertical, simplex, center-packed plunger pattern, one main and one auxiliary feed pump in each fireroom, operated from upper grating. The main and auxiliary feed pumps in each fireroom are the same size and identical in every respect except that zinc protectors are fitted in the auxiliary feed pumps only. The feed pumps in two-boiler or single firerooms are 9 inches by 6 inches by 12 inches; and those in four-boiler or double firerooms, are 11 inches by 8 inches by 12 inches. All feed pumps draw from fore-and-aft feed suction mains, one on each side of ship close to center-line bulkhead and well above fireroom floors, with a crossover connection in forward firerooms. The suction mains draw from an equalizing pipe between main feed tanks, with a connection to hot-well-pump discharge for maintaining a pressure at suction side of feed pumps. The main feed pump discharges through feed-water heaters or by-pass around heaters, to a fore-and-aft discharge main with crossover in forward double fireroom from which any one or all of the boilers in the ship may be fed. The auxiliary feed pumps discharge, independent of heaters, to any one or all of the boilers in its own compartment only. The auxiliary feed pump is also used as a fire and bilge pump, with all salt-water suctions led to a manifold connected to pump suction by a short length of piping.

Feed-Water Heaters.—There are eight feed-water heaters, of the high-pressure type, placed one in each fireroom on the discharge side of main feed pumps. The shell is cylindrical, of $\frac{3}{16}$-inch boiler plate, with flanges of angle iron turned to the form of a diaphragm to take up the shell expansion. The water chests are of cast steel and the heads of boiler plate bumped to a radius equal to diameter of shell. The tubes are of Muntz metal, $\frac{3}{4}$-inch outside diameter, No. 14 B.W.G., tinned inside and outside and expanded into tube sheets of 1-inch boiler plate. Each tube is fitted with a retarder consisting of a thin brass strip twisted and crimped to retard the

flow of feed water. The retarders are crimped over tube ends and held in place by perforated steel plates over tube sheets.

The heaters for single firerooms have 168 square feet of heating surface, and those for double firerooms have 323 square feet, and each one is piped to take the exhaust from the feed pumps and blowers in its own compartment and exhaust to auxiliary exhaust line. They drain into a drain-collecting main for engine and fireroom auxiliaries, with connections to main feed tanks and auxiliary condensers.

Reserve-Feed Tank.—That portion of inner bottom between frames 60 and 68½, beneath after fireroom and athwartship coal bunker, is divided into four compartments and used as reserve-feed tank with a total capacity of 112 tons. The hotwell pump in engine room is piped to supply and draw from these tanks, with a connection to a twin nozzle on either side of ship just beneath gun deck for taking aboard fresh water.

Main and Auxiliary Steam Piping.—The main steam pipes, of seamless-steel tubing, one on each side of ship near center of firerooms, is 5½ inches in diameter in forward fireroom, increasing in diameter at each connection to the auxiliary steam to a 12-inch diameter at engine-room bulkhead. The auxiliary steam pipe, running just outboard of and parallel to the main steam, is 8 inches in diameter throughout the firerooms, with a 5-inch branch to machinery forward and a 6-inch branch to engine rooms. The steam from each pair of boilers is admitted to the auxiliary steam line through a cross valve just beneath it, and in the auxiliary steam line in each fireroom there is a cross valve and cross connection to main steam line. At the forward engine-room bulkhead the main and auxiliary steam on the same side of ship, and the main steam lines on opposite sides of ship, are cross connected. The auxiliary steam lines on opposite sides of ship are cross connected in forward firerooms and after part of engine rooms. By this arrangement it is possible to isolate either the main or auxiliary steam pipes, or both, in

one or more firerooms without interfering with the passage of steam from the other boilers.

Throughout the firerooms the main and auxiliary steam lines are in straight leads, the expansion being provided for in slip joints anchored to protective-deck beams. In the engine rooms they are provided with expansion bends and slip joints at their connections to the main engine.

Ash Hoists.—There are eight 4½-inch by 4½-inch double-cylinder ash-hoisting engines of the Hyde Windlass Company's design, located one in each fireroom ventilator, with operating wheels in fireroom and at gun deck. They are fitted with follow-up gear and automatic stops to prevent over winding, and will hoist a bucket of ashes weighing 300 pounds from fireroom floor to gun deck, about 32 feet, in five seconds. At the gun deck the buckets are transferred to trolleys leading to ash chutes at side of ship.

Forced Draft Blowers.—There are twelve forced-draft blowers, two in each double fireroom and one in each single fireroom, of the upright type, with opposite cranks, designed by Bath Iron Works. They take air from the fireroom ventilators and discharge directly into the fireroom through fan casings. The two engines in the double firerooms are placed end to end and bed plates and shaft couplings bolted together.

Steam cylinders, two each, diameter, inches	5
Stroke, inches	5
Diameter of fan, inches	60
Area of induction nozzle, inches	1,133

ELECTRIC PLANT.

The generating plant consists of six 50- and two 100-kw General Electric Company's compound-wound generators, each direct connected to a cross-compound engine, located in four compartments just forward of machinery space, two being in the hold and two on platform just above. The 50-kw. generators are 6-pole, 400 ampères, 125 volts, designed for 400 revolutions per minute; the 100-kw. generators are 8-pole, 800 ampères, 125 volts, designed for 350 revolutions per minute.

The electrical equipment is practically the same as on the other vessels of this class which have been previously described in these pages. All power-driven appliances outside of machinery space, except windlass, steering engine and ice machine, are electrically operated.

Dynamo Condenser.—In the lower port-dynamo room there is a Wheeler Admiralty surface condenser with a Blake combined air and circulating pump attached. It has a cylindrical steel shell and composition water heads, tube sheets, glands and ferrules. The tubes are of composition tinned inside and outside. This condenser is also provided with a connection to the auxiliary exhaust, thus avoiding the shutting off the dynamos when repairs or overhauling of dynamo condenser or pumps becomes necessary.

Condenser, diameter, shell, outside, inches	32½
tubes, number of	864
cooling surface, square feet	1,200
Steam cylinder, diameter, inches	12
Air cylinder, diameter, inches	14
Circulating cylinder, diameter, inches	14
Stroke, inches	12

Data not given is the same as for auxiliary condensers.

MACHINE SHOP.

The engineer's workshop is located on a flat just beneath the protective deck, between engine rooms and after firerooms, and is divided into two compartments by the center-line bulkhead with a watertight door between. Entrance to workshop is obtained through a watertight door on upper grating in each engine room. All machine-shop tools are driven through line and counter shafting by a General Electric Company's 10-horsepower shunt-wound motor. The tools installed are as follows: One screw-cutting back-geared gap lathe, to swing 30 inches over the ways and to take 10 feet between centers; one 14-inch screw-cutting back-geared lathe, to take 4 feet between centers; one column-shaping machine of 15-inch stroke and 26-inch traverse; one back-geared, 28-inch, drilling machine; one 14-inch sensitive drill; one Brainard No. 14½

Universal milling machine; one combined hand punch and shears, with 6-inch shear blades, capable of cutting $\frac{3}{4}$-inch round iron, shearing $\frac{3}{8}$-inch steel plate, and punching $\frac{3}{8}$-inch holes in $\frac{3}{8}$-inch mild-steel plates; one emery grinder with two wheels 12 inches diameter and 2-inch face; one 30-inch grindstone; six bench vises.

Air Compressor.—In the engineer's workshop there is an 11-inch Westinghouse, locomotive-type, air pump and reservoir tank for maintaining an air pressure of 140 pounds per square inch, for use in the turret smoke-ejecting system. A branch from this pump is led fore and aft through machinery space, with a valve and hose connection in each compartment, for the use of the engineers in cleaning boiler tubes and driving pneumatic tools.

EVAPORATING PLANT.

The evaporating and distilling plant is located on berth deck, just forward of engine-room entrance, with all of its accessories in one compartment.

There are four Davidson "Type A" evaporators of 6,000 gallons each, and six Davidson distillers of 3,300 gallons each. The evaporator shells are of "Class B" boiler plate, $\frac{1}{2}$-inch thick, with heads bumped to a radius equal to diameter of shell, and without bracing. Each evaporator has eighty-four 2-inch brass tubes, in two nests of forty-two tubes each, with a total heating surface of 240 square feet. The distillers are of cast iron with composition tube sheets and expansion sleeves. Each distiller has 107 $\frac{5}{8}$-inch tinned-brass tubes, and a cooling surface of 66 square feet. The evaporators are piped for both single and double effect. All the pumps are of the Blake vertical, simplex, piston pattern.

There is one distiller circulating pump 12 inches by 14 inches by 12 inches with discharge to distillers and thence to flushing main or overboard or to flushing main direct. A connection to the fire main is also provided for distiller circulation. There are three $4\frac{1}{2}$-inch by 5-inch by 6-inch pumps used for evaporator feed, brine and fresh-water service. Dur-

ing a five-hours' test on July 5, 1906, the plant produced a mean of 1,236 gallons of potable water per hour, giving the plant an actual working capacity of 29,904 gallons per twenty-four hours.

REFRIGERATING PLANT.

On the berth deck forward there is an Allen dense-air ice machine with a cooling effect of three tons of ice per day. There are five cold-storage rooms and an ice-making tank adjacent to the ice machine, and cold-air pipes lead to these and to the two scuttle butts on gun deck forward.

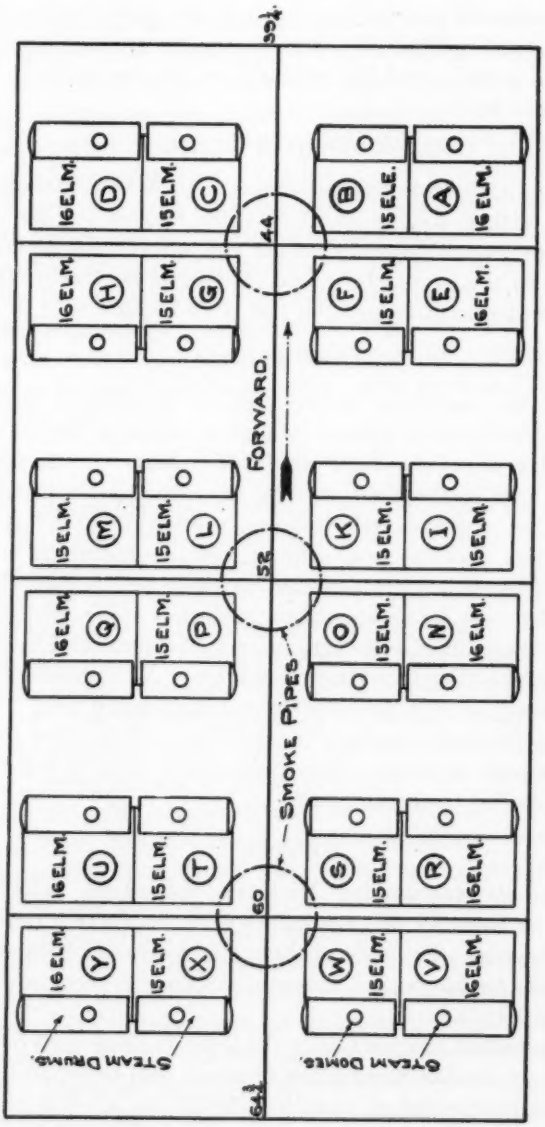
HEATING SYSTEM.

From the auxiliary steam pipe in evaporator room a 3-inch connection is taken, through a reducing valve, to a three-valve manifold, from which the steam is distributed to the three independent heating circuits. Each circuit is divided into groups, with valves so fitted that any group of radiators may be used without turning steam on entire circuit. There is a drain circuit corresponding to each steam circuit led to a float trap within machinery space, which discharges into a drain, collecting main with connections to main feed tank, auxiliary condenser and bilge. All radiators are 1-inch brass-pipe coils, no standing radiators being used. The steam to the galley cookers is taken from steam pipe to ice machine and windlass. The drains from pantries and bath heaters are led to separate traps.

BOILERS.

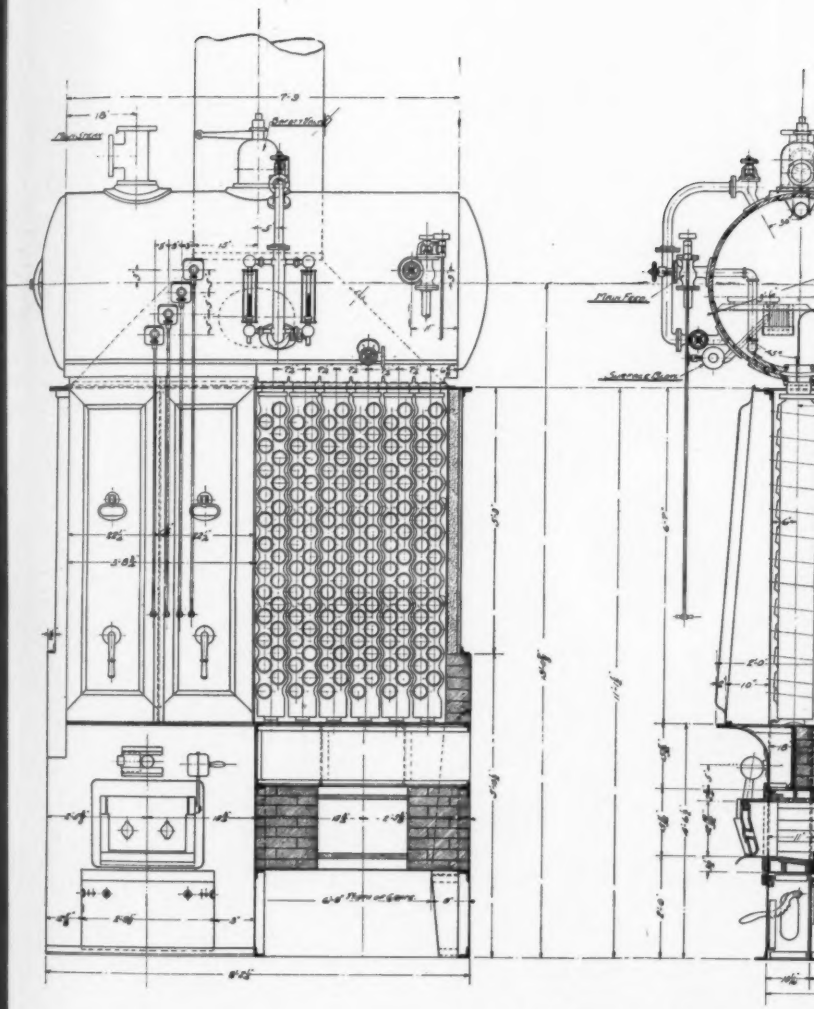
There are twenty-four Niclausse water-tube boilers arranged in batteries of two boilers each, in eight watertight compartments. Fourteen of these are 15-element boilers and ten are 16-element boilers, arranged as shown in accompanying diagram, Fig. 1.

These boilers are practically the same as those used on the *Virginia*, a detailed description of which will be found in the JOURNAL for November, 1905, Vol. XVII, No. 4. The trouble

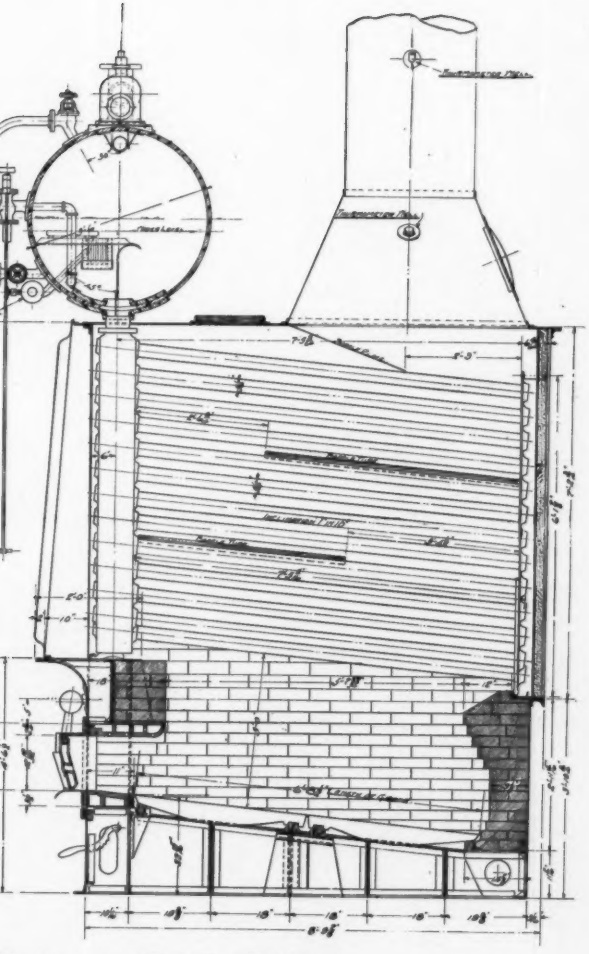


ARRANGEMENT OF FIREROOMS, U. S. S. "GEORGIA."

XU



NICLAUSSE BOILER AS INSTALLED ON



Note
Total Heating Surface = 1852.36 ft²
Total Grate Surface = 45.225 ft²

INSTALLED ON U. S. S. "GEORGIA."

caused by the excessive heating of boiler casings on the *Virginia* was eliminated in these boilers by a change in the firebrick and the addition of 1-inch block tile bolted inside of millboard and asbestos and extending from top of firebrick to top of casing.

BOILER DATA.

Number of boilers	24
15-element boilers	14
16-element boilers	10
tubes to element	24
Generating tubes, diameter, inches	03¼
thickness of	B.W.G., No. 6
length of, exposed, feet and inches	7–05$\frac{11}{16}$
Circulating tubes, diameter outside, inches	01½
Heating surface, one 16-element boiler, square feet	2,474.44
15-element boiler, square feet	2,320.07
all boilers, square feet	57,225.45
Grate surface, one 16-element boiler, square feet	62.15
15-element boiler, square feet	57.93
all boilers, square feet	1,432.52
Ratio of total G.S. to total H.S.	39.95
Weight of 16-element boiler with water, pounds	69,974
without water, pounds	60,075
15-element boiler with water, pounds	66,057
without water, pounds	56,774
Steam pressure, pounds, per square inch	265

OFFICIAL TRIAL.

Standardization of Screws.—The vessel was tried by the standardized-screw method, progressive runs being made over the measured-mile course off Rockland, Maine, June 13, 1906.

The data recorded in Tables I and II were obtained, from which the curves shown on Plate I were plotted.

The draught and corresponding displacement at the beginning and end of runs were as follows:

	Beginning.	End.
Draught, forward, feet and inches, . .	23–09$\frac{1}{8}$	23–06$\frac{3}{8}$
aft, feet and inches, ...	23–09$\frac{3}{8}$	23–06$\frac{1}{4}$
Displacement, tons, ...	15,003	14,810

The results of the runs with and against the tide having been plotted as separate curves, the curve of true speed was obtained, from which it was deduced that a mean of both

Table I.—U. S. S. *GEORGIA.*

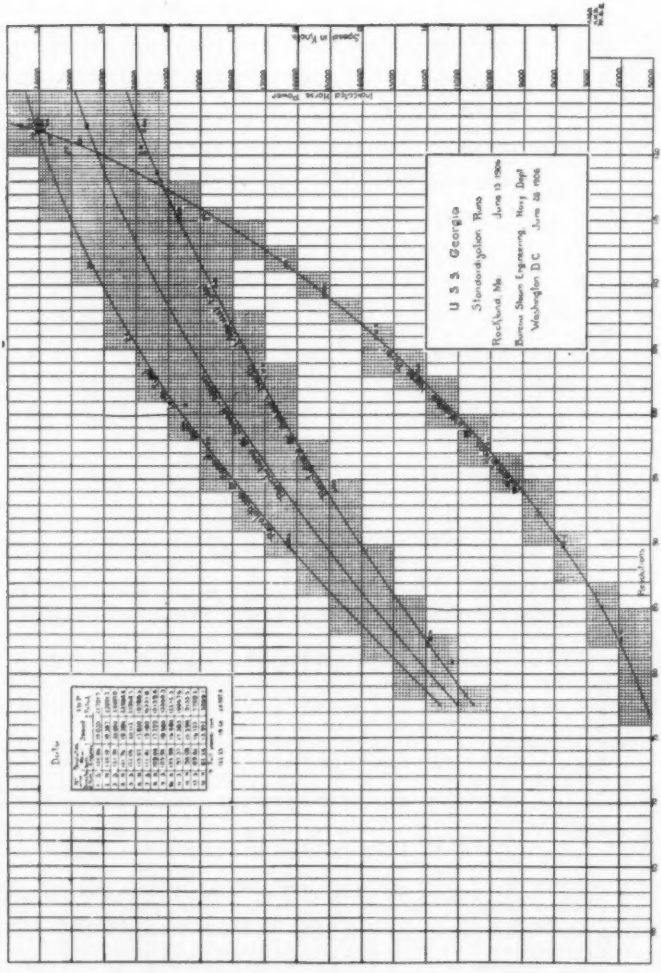
Data from Progressive Runs over Measured-Mile Course off Rockland, Me., June 13, 1906.

Number and direction of run.	Time on course.	Revolutions of main engines.					Speed in knots per hour.	Indicated horsepower.											
		To make 1 knot.		Per minute.				Starboard engine.					Port engine.					Total of both main engines.	Auxiliaries in operation.	Total for all machinery.
		Starboard.	Port.	Starboard.	Port.	Mean.		H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.	H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.			
	m. s.																			
1, S.	3-01.5	365.6	365.4	120.9	120.8	120.9	19.83	2,800.9	4,086.0	1,969.0	2,115.0	10,970.9	2,394.1	4,847.7	2,185.7	2,303.3	1,730.8	22,701.7	643.3	23,345.0
2, N.	3-16.0	393.1	393.0	120.2	120.2	120.2	18.37	2,348.0	4,579.0	2,097.0	2,212.0	11,236.0	1,710.3	5,085.7	2,511.0	2,548.3	11,855.3	23,091.3	706.8	23,798.1
3, S.	2-59.5	364.5	364.7	121.9	122.0	121.9	20.05	2,423.0	4,870.0	2,107.0	2,275.0	11,675.0	1,925.1	5,454 0	2,438.2	2,516.5	12,333.8	24,008.8	706.8	24,715.6
4, N.	3-15.8	397.1	397.2	121.7	121.8	121.7	18.39	2,282.0	5,013.0	2,184.0	2,341.0	11,820.0	1,775.0	5,221 4	2,766.1	2,401.9	12,164.4	23,984.4	695.5	24,680.9
5, S.	2-59.0	363.6	364.7	121.9	122.2	122.1	20.11	2,630.0	4,531.0	2,186 0	2,303.0	11,650.0	1,781.8	5,466.1	2,494.1	2,572.5	12,314.5	23,964 5	690.5	24,655.0
6, N.	3-21.8	387.8	387.0	115.2	114.9	115.1	17.84	3,283.0	2,867.0	1,381.0	1,530.0	9,061.0	3,337.8	3,238.8	1,519.1	1,631.6	9,727.3	18,788.3	471.6	19,259.9
7, S.	3-07.7	349 2	348.0	111.6	111.2	111.4	19.18	2,940.0	2,551.0	1,200.0	1,317 0	8,008.0	2,862.0	2,826.8	1,295.2	1,339.8	8,323.8	16,331.8	428.8	16,760.6
8, N.	3-28.1	377.8	378.6	108.9	109.2	109.0	17.30	2,660.0	2,376.0	1,063.0	1,150 0	7,249.0	2,759.5	2,607.6	1,190.1	1,333.2	7,890.4	15,139 4	327 7	15,467.1
9, S.	3-13.7	341.8	343.1	105.7	106.1	105.9	18.58	2,478.0	1,975.0	970.0	1,065.0	6,488.0	2,573 7	2,339.8	1,060.5	1,103.3	7,077 3	13,565.3	345.6	13,910.9
10, N.	3-38.4	375.0	374.8	103 0	102.9	103.0	16.48	2,246 0	1,798.0	852.0	918.0	5,814.0	2,309.7	2,150.5	952.5	985.6	6,398.3	12,212.3	292.7	12,505.0
11, S.	3-27.1	337.1	335.9	97.5	97.2	97.4	17.38	1,939.0	1,498.0	702.0	749.0	4,888 0	1,837.1	1,785.9	770.6	786.0	5,179.6	10,067.6	284.0	10,351.6
12, N.	3-53.8	366.6	366.3	94.1	94.0	94.1	15.40	1,700 0	1,365.0	591.0	676.0	4,332.0	1,799.0	1,607 0	699.1	698.4	4,803.5	9,135.5	266.4	9,401.9
13, S.	3-43.1	333.7	333.8	89.8	89.8	89.8	16.13	1,530.0	1,175 0	505.0	548.0	3,758.0	1,493.4	1,337.0	578.7	541.1	3,950.2	7,708.2	262 7	7,970 9
14, N.	4-17.9	353.0	353.4	82.2	82.3	82.2	13.96	1,141.0	879.0	394.0	380.0	2,794.0	1,252.6	1,108.9	455.4	448.2	3,165.1	5,959.1	253.6	6,212.7

Table II.—U. S. S. *GEORGIA.*

Data from Progressive Runs over Measured-Mile Course, off Rockland, Me., June 13, 1906.

Number and direction of run.	Which engine.	Steam pressures.			Mean effective pressures.				Vacuum, in inches.	Air pressure, in inches of water.	Cut-off of cylinders from beginning.				Temperatures.						Revolutions or double strokes.								
		In boilers above atmosphere.	At engine above atmosphere.	H.P. steam chest, absolute.	In sat receiver, absolute.	In sd receiver, absolute.	H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Referred to L.P.		H.P.	I.P.	Forward L.P.	Aft L.P.	Outside air.	Engine-room working platform.	Fireroom working level.	Injection.	Discharge.	Hotwell.	Feed.	Main feed pump.	Auxiliary feed pump.	Hotwell pump.	Air pump.	Bilge pump.	Dynamo condenser pump.	Sanitary pump.	Circulating pump.	Dynamo.	Fireroom blowers.	Pyrometer, degrees, 49 feet above grate.
1, S.	S. P.	278	260	265	110	40	102	55	19.7	21.2	54.93	26	2.02	Full.			50	75	80	50	116	84	176	43	27	34	25	37	13	35	164	350	681	
2, N.	S. P.	271	250	263	138	44	86	55.5	21.9	23.1	38.80	26	2.05				52	75	81	51	118	95	173	43	27	32	14	41			178		681	
	P.		258	271	127	41	86	68	31.1	23.3	36.58	26									116	92		38	31	34	18	37			200		656	
3, S.	S. P.	280	263	278	156	44	86.5	65	30.1	25.7	59.75	26	2.02	Full.			52	75	79	52	110	95	173	38	31	31	23	42	13	35	178	350	656	
	P.		265	278	160	43.5	87.5	68	30.9	28.6	57.96	26									116	94		38	30	34	24	31			225		641	
4, N.	S. P.	279	263	278	145	46	89.3	73	34.2	29.0	61.24	25.7	2.02				53	75	80	50	116	94	181	38	30	39	21	34	13	35	168	350	641	
	P.		267	279	147	44	92.5	77	31.7	29.3	62.76	26.5	2.04				54	75	84	51	113	92	180	38	29	34	25	31	13	35	225	330	624	
5, S.	S. P.	282	265	282	146	47	94	80	31.7	29.9	63.30	25.5	2.04				54	76	84	51	115	98	180	38	29	33	20	33			200	330	624	
	P.		267	282	147	43.5	95	84	31.7	29.9	63.87	26	2.06				56	76	84	51	98	88	175	38	29	33	27	16	13	35	197	330	623	
6, N.	S. P.	279	271	279	150	47	96	82	34.7	29.5	61.01	26						75	77	51	110	99	175	38	32	35	29	33	13	35	208	350	623	
	P.		270	279	154	47	98.5	84	32.5	30.1	62.63	26.5	2.06				57	75	87	51	94	69	171	36	28	30	19	30			161		483	
7, S.	S. P.	279	272	279	160	48	98	84	32.5	30.1	63.25	26.5	1.86				57	75	78	51	94	90	171	36	28	37	32	40	13	35	200	350	483	710
	P.		276	280	160	48	98	84	32.5	30.1	63.25	26.5	1.86				58	75	76	50	100	90	171	36	28	30	10	31	13	35	200	350	483	
8, N.	S. P.	276	280	280	160	48	98	84	32.5	30.1	63.25	26.5	1.00				58	75	76	50	97	90	153	37	24	40	20	44	13	35	196	350	431	
	P.		272	276	160	48	98	84	32.5	30.1	63.25	26.5									104	90		37	24	36	10	31			175		431	
9, S.	S. P.	281	275	281	160	48	98	84	32.5	30.1	63.25	26.5	0.52				58	75	77	52	95	90	154	39	20	28	20	17	13	35	174	350	680	
	P.		275	281	160	48	98	84	32.5	30.1	63.25	26.5									103	90		39	20	28	19	30			156		321	
10, N.	S. P.	272	275	278	160	48	98	84	32.5	30.1	63.25	26.5	0.44				65	75	95	50	95	90	160	39	20	18	9	31	13	35	175	350	321	
	P.		270	272	160	48	98	84	32.5	30.1	63.25	26.5									103	90		39	20	24	19	32			160		344	
11, S.	S. P.	270	270	270	160	48	98	84	32.5	30.1	63.25	26.5	0.14				67	75	96	51	92	57	154	25	16	10	9	19	13	19	144	350	344	
	P.		270	270	160	48	98	84	32.5	30.1	63.25	26.5									108	68		25	16	26.5	19	23			169		239	
12, N.	S. P.	274	270	270	160	48	98	84	32.5	30.1	63.25	26.5	...				68	75	98	50	84	56	166	22	15	10	10	20	13	19	144	350	239	
	P.		270	274	160	48	98	84	32.5	30.1	63.25	26.5									88	55		22	15	10	20	25			169		234	
13, N.	S. P.	274	270	274	173	51	103.2	79.5	37.6	31.1	67.86	26.5	...				69	74	99	50	93	55	150	22	13	11	7	12	13	22	153	350	234	
	P.		270	270	172	51	103.2	84	37.6	31.1	67.86	26.3									91	60		22	13	12	19	35			152		248	540
13, S.	S. P.	249	248	248	155	47	75	21.3	6.8	7.4	25.33	27	...				72	74	101	50	80	56	153	22	12	9	9	19	13	21	152	350	238	
	P.		248	249	145	50	17	24.3	7.8	7.3	26.63	26.3									84	58		22	12	16	18	28			140		238	
14, N.	S. P.	275	275	275	132	40	15	17.4	5.8	5.6	20.57	27	...				73	74	101	51	54	54	158	17	9	10	9	17	13	23	152	350	264	
	P.		265	275	132	45	15	22	6.7	6.6	23.89	26.3									80	55		17	9	12	19	29			152		264	


U S S Georgia
Standardization Runs
Rockland Me June 13 1906
Bureau Steam Engineering Navy Dept
Washington D C June 18 1906
Indicated Horse Power
Speed in Knots
Data
Plate I.

engines of 118.9 revolutions per minute would be required to give the contract speed of 19 knots.

The coal used was Pocahontas run-of-the-mine.

OFFICIAL FOUR-HOURS' TRIAL.

On the morning of June 14, 1906, the *Georgia* got underway and went to sea for the four-hours' official trials.

The draught and displacement at the beginning of the trial were—

Draught, forward, feet and inches, ... 23–10¼
aft, feet and inches, ... 23–10½
mean, feet and inches, ... 23–10⅜
Corresponding displacement, tons, ... 15,070

The trial commenced at 5h. 46m. A. M., June 14, 1906, and finished at 9h. 46m. A. M. The weather was clear and pleasant, with light breezes from N. W. and a smooth sea.

Average revolutions per minute, ... 122.25
Corresponding speed from curve, knots, ... 19.26

A synopsis of the data obtained follows:

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

(*Average of one half-hourly observations.*)

STEAM PRESSURES.

	Starboard.	*Port.*
Steam pressure at boilers (per gauge), pounds	261.9	
H.P. steam chest (gauge), pounds	231.0	232.0
1st receiver (absolute), pounds	128.0	135.0
2d receiver (absolute), pounds	51.0	54.0
Vacuum in condensers, inches of mercury	25.5	25.2

TEMPERATURES.

Injection, degrees	50.0	50.0
Discharge, degrees	107.0	111.7
Hotwell, degrees	88.0	90.3
Feed-water, degrees	158.0	
Engine room, upper platform, degrees	94.0	96.1
working platform, degrees	80.0	91.4
Firerooms, above gratings, degrees	89.9	
working level, degrees	88.7	
Smoke stacks, average	789.0	

Plate I.

REVOLUTIONS, OR DOUBLE STROKE, PER MINUTE.

	Starboard.		*Port.*
Main engines (mean for trial)	122.44		122.06
average for both		122.25	
Pumps, main air	26.4		24.3
circulating	257.0		251.6
feed		42.0	
hotwell	38.5		38.8
bilge	27.7		44.9
sanitary		26.6	
auxiliary, feed	...		...
condenser	...		...
Blower engines		605.0	
Speed of ship, in knots per hour		19.26	
Slip of propeller, in per cent. of its own speed based on mean pitch and average speed	17.33		16.81
Air pressure, in firerooms, inches of water		1.84	

MEAN EFFECTIVE PRESSURES IN CYLINDERS, IN POUNDS PER SQUARE INCH.

Main engines, H.P. cylinder	71.0	64.5
I.P. cylinder	54.5	57.0
F.L.P. cylinder	29.4	32.38
A.L.P. cylinder	29.8	31.81
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	59.67	62.15

INDICATED HORSEPOWER.

	Starboard.	*Port.*
Main engines, H.P. cylinder	1,975.0	1,794.8
I.P. cylinder	4,105.0	4,263.1
F.L.P. cylinder	29,81.0	3,264.1
A.L.P. cylinder	3,010.0	3,204.4
Total	12,071.0	12,526.4
both engines	**24,597.4**	

AUXILIARIES.

Air pumps	18.6		17.8
Circulating pumps, main	106.0		85.0
Feed pumps (main and auxiliary)		263.2	
Hotwell pumps	7.55		7.6
Bilge pumps	1.6		2.75
Sanitary pump	3.5		0.0
Auxiliary condenser pumps	...		...
Blower engines (forced draft)		312.0	
Steering engine (estimated)	...		...
Dynamo engine (one in operation)		40.2	
Total		**865.8**	
all machinery in operation		**25,463.2**	

COAL.

Kind and quality	Pocahontas, hand picked, excellent.
Pounds per hour	62,288.0

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	17.5
heating surface	0.445
Pounds of coal per I.H.P. per hour (main engines only)	2.53
all machinery in operation	2.446
Pounds of coal per square foot of grate surface	43.49
heating surface	1.088
Cooling surface (main condenser), square feet per (total) I.H.P.	0.93

The working of the machinery, both main and auxiliary, and its performance during the trials was satisfactory. The boilers steamed freely and worked in a satisfactory manner throughout the trials. A careful observation and inspection of the various parts of the machinery during and after the trials showed the engines, boilers and appurtenances to be in excellent condition.

MODERN METHODS OF PIPE FLANGING BY MACHINERY.

By Luther D. Lovekin, Associate.

The subject of attaching flanges to pipes for all purposes is one to which the writer has given considerable thought and attention during the past six years, and, having investigated the various methods in use, begs to submit to the engineering world a new method, which has been well tried, and found to possess great advantages over all other methods known to the writer.

One of the chief advantages in connection with this work is its application in attaching flanges to copper pipe, principally because the practice in this field of work has remained almost constant during the past fifty years or more.

Thousands of dollars are unnecessarily expended annually owing to the crude methods employed in doing this work by hand.

The following is a brief description of the present methods employed in most copper shops throughout the country, together with a comparison of the work necessary when using the Lovekin machines and tools in order to produce work of the character shown in Figures Nos. 5 and 6 of this article, viz.

In order to connect the flange and pipe under the old methods, flanges are made and *turned on the face and bored,* after which the copper pipe is entered into the same, and pened or hammered by hand, so much as to distort the flange in almost every case. This, however, does not form a tight joint, and in order to prevent leakage, brazing has to be resorted to, which, as is well known, materially decreases the strength of the pipe, inasmuch as excessive heating destroys the properties of the copper and warps and twists the flanges.

After this is finished another expense is added. The pipe is then taken to the machine shop and placed on a boring and facing machine, to be faced up, and the amount of time necessary to reface this flange complete depends entirely on the shape of the pipe and the size of same, a 4-inch pipe requiring anywhere from one to two hours to complete said work. This amounts to several thousand dollars a year in large shipbuilding plants, and when finished the character of the work will not begin to compare with that done by the machine.

With the machine-made joint the flange is bored and faced, placed in the machine, pipe entered and clamped, and the operation completed in about ten minutes for a 4-inch pipe, the expanding, flaring and facing being perfect; and, inasmuch as we can extrude the metal into recesses made in the flanges, we know the danger of leaking is positively less than that of boiler tubes, where we resort to expanding into holes simply bored in the tube plate.

I believe the old rules compelling the brazing of pipes were originated on account of the impossibility of securing tight joints with high pressure, leakage occurring between pipe and flange on the outside; but with tests which we have conducted with our machine-made joints, before the U. S. Steamboat Inspectors and the U. S. Navy Department, brazing is not essential, and there is no comparison as to neatness, efficiency and cost of production.

The Lovekin machines and tools were designed to meet the demands of shipbuilders and engineers generally where a large amount of wrought-iron, steel and copper piping is required to be expanded tightly in the flange, as well as flared at the ends, thus dispensing with the common practice of threading, rivetting or peneing by hammer, especially so when such pipes are subjected to the action of salt water or to any vibration.

Steam and exhaust piping, and also the bilge and ballast systems of vessels, have given considerable trouble due to using the threaded pipe, and therefore it is about time such practices were discontinued.

I will now describe the new list of machines and tools which are being placed on the market to do this class of work in the most approved manner, with a saving that should startle the proprietors of all such establishments where these old methods prevail.

DESCRIPTION OF CLASS "A" LOVEKIN EXPANDING AND FLANGING MACHINE FOR COPPER PIPES, 2 INCHES TO 7½ INCHES INCLUSIVE, AND STANDARD LAP-WELDED STEEL PIPE UP TO 6 INCHES DIAMETER.

See Figures Nos. 1, 2, and 3.

This machine consists of a cast-iron base, having at the rear end a motor for driving the same. (See Fig. 2.) The motor drives the machine by means of a belt to the cone pulleys, so as to permit of variable speeds being used.

The cone-pulley shaft is provided with a worm which engages in a worm wheel suitably housed on the rear end of the machine. The worm wheel has a long sleeve cast on the same, and is also provided with a round collar or nut at the extreme end of this sleeve. This worm wheel and its sleeve is provided with a key-way, so as to allow the central or driven shaft to move freely forward and backward as desired, and the feather transmits the power to the central or driven shaft, causing it to revolve.

The worm and worm wheel, together with their brackets, form a fixed part of the machine with relation to the driving mechanism. The central shaft passes through the worm wheel, and is enlarged at the front end to form a socket for all kinds of tools. Between this enlarged portion and the bracket "B" suitable anti-friction rings are provided, thus reducing the friction to a minimum. This shaft is secured at the rear end of the movable bracket "B" by means of a round nut, as shown in Fig. 1. This enables the shaft to revolve freely in this movable bearing, but it cannot move forward unless bracket "B" is advanced.

The lead screw for moving the bracket "B" to and from the

work to be operated upon, is provided with bevel wheels of suitable size, and operated by a hand wheel, located to suit the best position of the operator in charge.

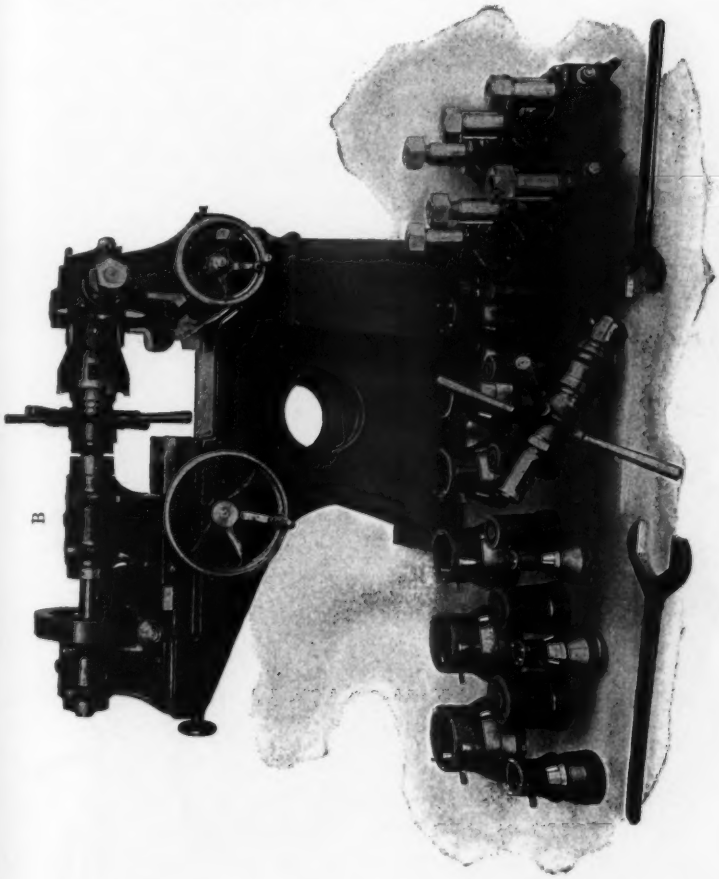


Fig. 1.

In addition, the central shaft revolves in this bracket; and to provide for the end thrust, which tends to lift the bracket upward, at the forward end, the lower edges are provided with good and substantial guide strips.

The front portion of the machine consists of two sliding heads mounted on suitable guideways on the base of the machine. These heads have a right-and-left-hand screw passing

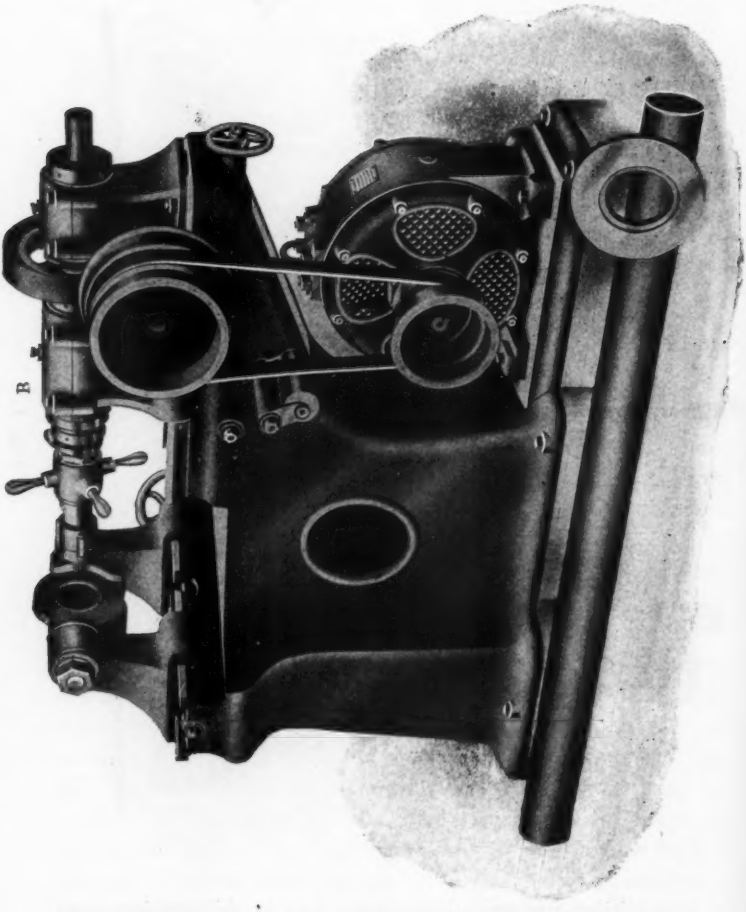


Fig. 2.

through nuts on the under side of the same, causing said heads to move from each other at equal distances from the center. The heads are made of cast iron, and are fitted with detachable

Fig. 2.

Fig. 3.

FIVE CLASS "A" MACHINES DELIVERED TO VARIOUS U. S. NAVY YARDS DURING THE YEAR OF 1905.

clamping jaws of cast iron. These jaws are also provided with steel "V" grips, for holding the pipe centrally and to prevent pipe from turning while being expanded.

The cast-iron jaws are made with grooves of suitable size to permit the flange being held in the same independent of the pipe. These grooves are faced, so as to be in perfect alignment with the central shaft. This permits the flange being placed exactly in line as desired, and at the back end suitable set screws are located so as to force the faced portion of the flange against the faced portion of the jaws. The backs of flange being rough castings, and at times having bosses on the back, makes this essential.

The machine is also provided with suitable locker and shelves at the front end. These will be found useful for stowing small parts.

DESCRIPTION OF TOOLS USED IN CONNECTION WITH THE LOVEKIN EXPANDING MACHINE—(CLASS "A").

The tools consist of a steel casting, shank and body in one piece, bored out to receive a forged-steel mandrel, as shown in Figs. 4 and 4-A.

The body is slotted out to suit the feeding screw "F", in order to give the inner mandrel "E" the proper movement necessary for expanding. The feeding screw "F" engages into the feeding wheel "G", and, as will be seen, any movement of wheel "G" causes the feed screw "F" to move outward or inward as desired. Feeding wheel "G" is held in place by a nut at the rear end; this nut also takes the thrust due to forcing the expanding and flaring rollers ahead, and is also provided with a facing at the rear end, which, when unscrewing from the mandrel proper, comes in contact with the socket head on machine, and thus makes a very useful device for withdrawing the tool shank. The shank is provided with suitable feather and set screw for holding the tool in place, and the end of shank comes in contact with the bottom of the bore in socket, thus giving a very firm backing for the tool proper in case of excessive end thrust.

As will be seen, the shank is arranged at its front end to

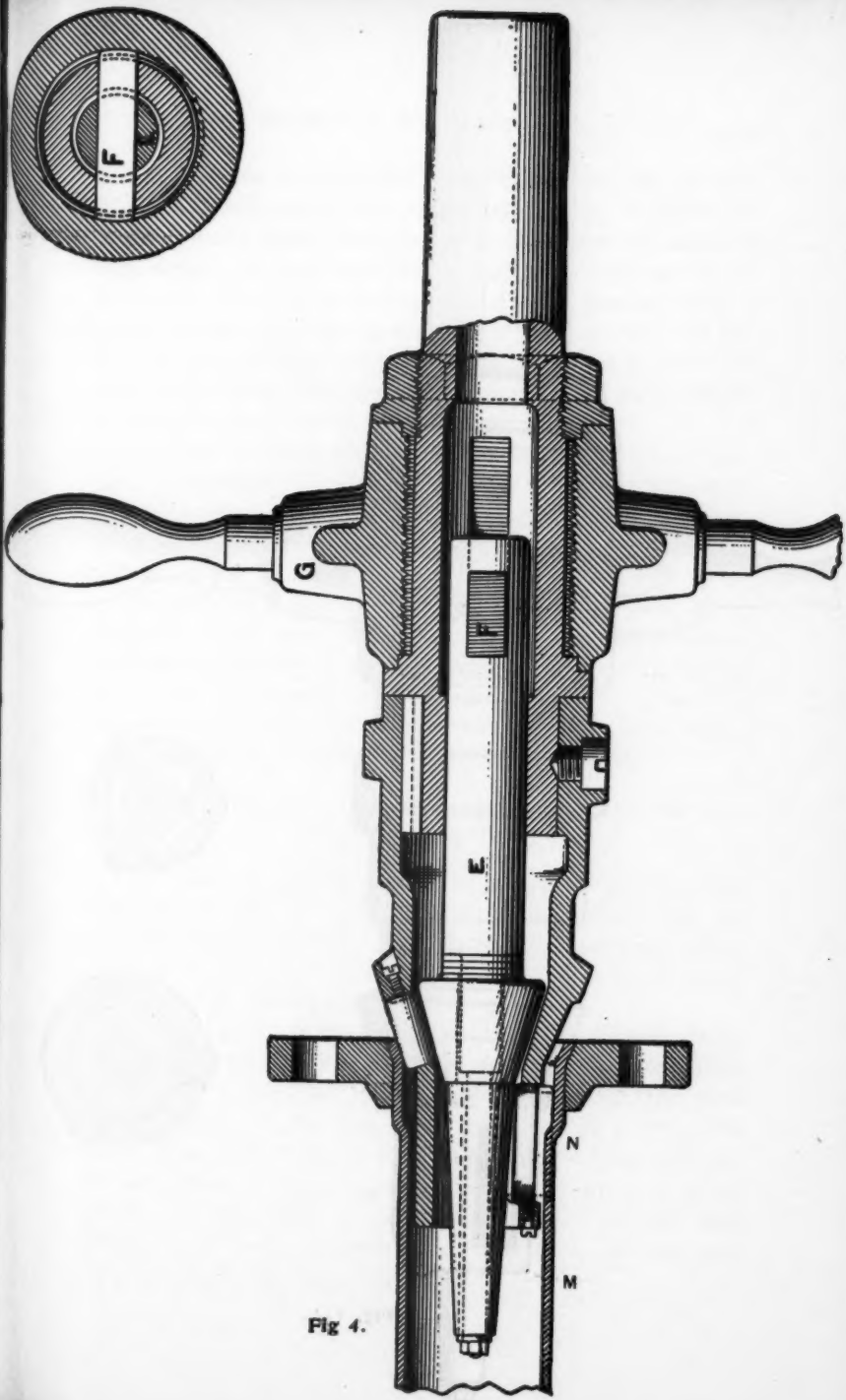


Fig 4.

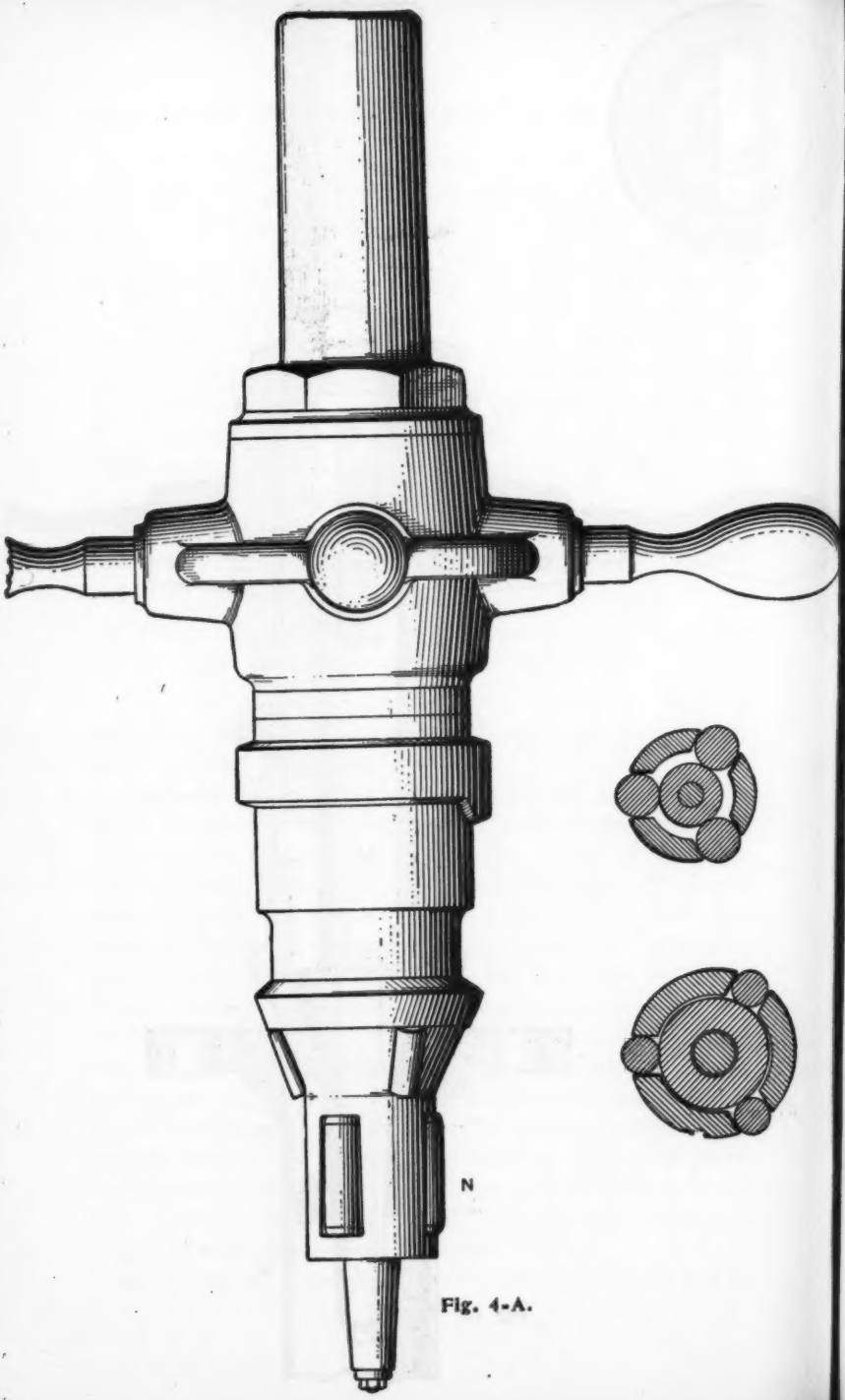


Fig. 4-A.

receive a series of detachable heads to suit the various sizes of pipes. These heads are made of steel castings, bored to suit the end of the shank on body of tool. They are also provided with suitable key-way and feather, in addition to the set screw, as shown. Each head has a series of slots, suitably finished to receive the expanding and flaring rollers, and the said rollers are held in place by means of set screws, as shown. All faces on the heads coming in contact with rollers are hardened, to insure lasting qualities.

The central rollers are mounted freely on the reduced portion of mandrel "E", and are provided with suitable anti-friction washers at their rear end to absorb the thrust. These rollers are steel, hardened, and are held in place by suitable nut and washer at the end of the mandrel, as shown.

The principle embodied in this expander is that of a central revolving roller, upon which operate the expanding rollers. The mandrel having no work to perform further than forcing the central rollers outward and inward as desired, is therefore not subjected to the extreme torsional stresses of other types, which constantly require new mandrels being made.

DIRECTIONS FOR OPERATING THE LOVEKIN EXPANDING MACHINE—(CLASS "A").

First, place the pipe and flange in jaws provided for same; then advance the tool so that the expanding rollers enter the pipe for about one inch; then force the expanding rollers outward by means of the hand-feeding wheel so as to insure the pipe being central with the tool.

While the pipe is in this central position with the tool, clamp flange at the back by means of the set screws provided; then set the "V" grips to the outside diameter of the pipe (being careful to see that these grips are in proper contact); then clamp the grips while in this position, and advance the set screws at the rear end of these "V" grips so as to prevent any possibility of the grips slipping. The pipe and flange are now held independently of each other. We are now ready to begin the expanding.

In order to expand, advance tool so that the end of the expanding rollers are at a distance of about one-quarter inch more than the total depth of the flange at the bore. This is done to insure the expanding showing up clearly at the back of the flange. While the tool is in this position, start motor, which will revolve the tool. The tool, in revolving, however, does not work on the pipe until we grasp the handles, which are in connection with the internal feeding screw, causing a retarding or breaking action, thereby forcing the central

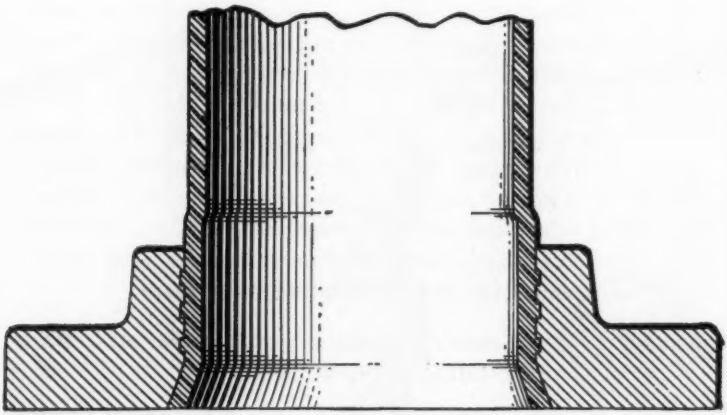


Fig. 5.

mandrel, with its conical rollers, ahead, which causes the expanding rollers to move outward.

The amount of retarding necessary to completely expand the pipe depends on various conditions, and therefore must be governed entirely by the operator in charge.

After pipe is expanded reverse the direction of the feeding wheel, which will cause the expanding rollers to collapse; withdraw the tool from the pipe, and remove the central roller "M", or preferably the three small expanding rollers "N". Then advance the tool until the flaring rollers are sufficiently within the flange, and continue the retarding or breaking

action in a similar manner to that of expanding until the pipe is completely flared. Withdraw tool again, and insert facing cutter; then advance the tool so as to trim off the ends of the pipe flush with the flange.

Fig. 6.

When accustomed to this work, the operation of placing the flange and pipe in the machine, expanding, flaring and facing has been done on many occasions with 3½ inch steel pipe, ⅛ inch thick, in 7½ minutes.

A table giving the amount of time taken to expand, flare and face pipe by the Lovekin machine is appended.

In conclusion I would state that Lovekin methods can be

used to advantage on locomotive boilers (no stay tubes required; every tube acts as a stay tube by our methods), evaporators (brass or steel tubes), distillers (brass or steel tubes), feed-water heaters (brass or steel tubes), marine boilers (no stay tubes required; every tube acts as a stay tube by our methods), brass pipe (for all purposes), copper pipe (for both high and low pressures, steam or water), all classes of iron and steel pipe (for both high and low pressures, steam or water), ammonia work of all kinds, steam-heating and ventilating-pipe work, particularly all high-pressure work of any character or for any purpose.

Special loose-flange pipe joints are made by our process, thus giving us every advantage over our competitors in high-pressure steam pipes for large power houses, as well as the highest grade of marine work.

Every engineer knows the great advantage of a loose-flange joint when erecting large piping, and European engineers are using these methods almost entirely.

These machines are made in three sizes, which are classed as follows: Class "A", for pipes 2 inches to 7½ inches in copper, 2 inches to 6 inches in steel; class "B", for pipes 5½ inches to 16 inches in copper, iron or steel, up to ½ inch thickness; class "C", for pipes 12 inches to 30 inches in copper, iron or steel, up to ½ inch thickness.

Descriptions of these machines may be had upon application to The International Engineering Company, 719 Pennsylvania Building, Philadelphia, Pa., U. S. A., who control the patents in all of the countries except U. S. and Canada. Or the Lovekin Pipe Expanding and Flanging Machine Company, No. 421 Chestnut St., Philadelphia, Pa., U. S. A., for the countries of United States and Canada.

List giving the average time taken to expand and flare into flange, and face off the ends of iron or steel pipe by the Lovekin Expanding and Flanging Machine. Flanges are furnished bored and faced, complete, time includes handling of pipe and setting flange.

CLASS "A" MACHINE, (One man only—No helper.)

Diameter of pipe.	*Thickness of material, iron or steel*	*Average time, including setting.*
2 inches.	$\frac{1}{4}$ to $\frac{5}{16}$ inch.	8 minutes.
$2\frac{1}{2}$	$\frac{1}{4}$ to $\frac{5}{16}$	9
3	$\frac{1}{4}$ to $\frac{5}{16}$	10
$3\frac{1}{2}$	$\frac{1}{4}$ to $\frac{1}{2}$	11
4	$\frac{1}{4}$	12
$4\frac{1}{2}$	$\frac{1}{4}$	14
5	$\frac{1}{4}$	16
$5\frac{1}{2}$	$\frac{1}{4}$	18
6	$\frac{1}{4}$	20
$6\frac{1}{2}$	$\frac{1}{4}$	22
7	$\frac{1}{4}$	24
$7\frac{1}{2}$	$\frac{1}{4}$	26

CLASS "B" MACHINE, (One man and helper.)

$5\frac{1}{2}$ inches.	$\frac{1}{4}$ to $\frac{3}{8}$ inch.	15 minutes.
6	$\frac{1}{4}$ to $\frac{3}{8}$	16
$6\frac{1}{2}$	$\frac{1}{4}$ to $\frac{3}{8}$	17
7	$\frac{1}{4}$ to $\frac{3}{8}$	18
$7\frac{1}{2}$	$\frac{1}{4}$ to $\frac{3}{8}$	19
8	$\frac{1}{4}$ to $\frac{3}{8}$	20
9	$\frac{1}{4}$ to $\frac{3}{8}$	21
10	$\frac{1}{4}$ to $\frac{3}{8}$	22
11	$\frac{1}{4}$ to $\frac{3}{8}$	23
12	$\frac{1}{4}$ to $\frac{1}{2}$	25
14	$\frac{1}{4}$ to $\frac{1}{2}$	27
16	$\frac{1}{4}$ to $\frac{1}{2}$	30

CLASS "C" MACHINE, (One man and helper.)

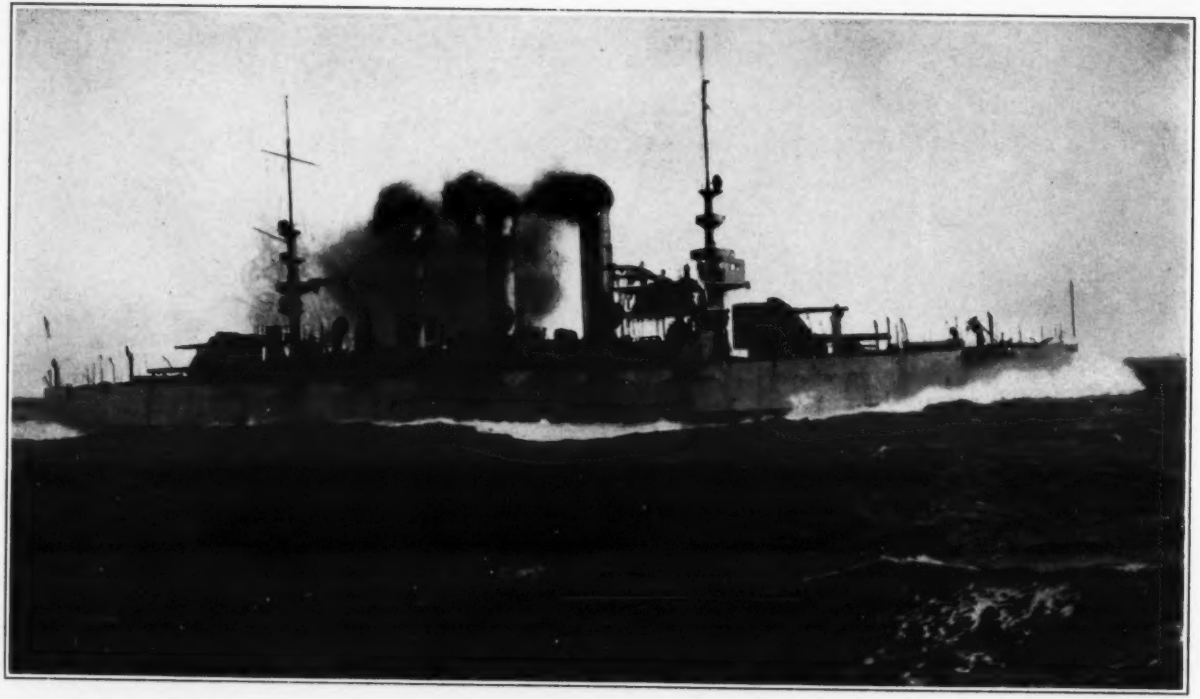
12 inches.	$\frac{3}{8}$ to $\frac{1}{2}$ inch.	25 minutes.
14	$\frac{3}{8}$ to $\frac{1}{2}$	27
16	$\frac{3}{8}$ to $\frac{1}{2}$	30
18	$\frac{3}{8}$ to $\frac{1}{2}$	35
20	$\frac{3}{8}$ to $\frac{1}{2}$	37
22	$\frac{3}{8}$ to $\frac{1}{2}$	40
24	$\frac{3}{8}$ to $\frac{1}{2}$	45
26	$\frac{3}{8}$ to $\frac{1}{2}$	50
28	$\frac{3}{8}$ to $\frac{1}{2}$	55
30	$\frac{3}{8}$ to $\frac{1}{2}$	60

Copper pipe requires one-third less time; larger sizes than those given are special; thickness of material over that given above is special.

One of these Class A machines has been in continual use at the works of The New York Shipbuilding Company for a

period of over two years, and during that time has expanded fully 20,000 flanges in the most perfect manner for both U. S. naval vessels as well as the merchant marine.

The U. S. Navy Department has ordered six Class A machines for use in their various Navy Yards.



U. S. Battleship "New Jersey."

U. S. BATTLESHIP *NEW JERSEY.*

DESCRIPTIONS—OFFICIAL TRIAL.

The *New Jersey* is a twin-screw armored battleship built by the Fore River Shipbuilding Company of Quincy, Mass. The contract was signed February 15th, 1901, the contract price being $3,405,000, of which $1,123,650 was for machinery. This price does not include the armor and armor bolts (exclusive of the protective deck), the ordnance and ordnance outfit and certain articles supplied by the Government, but does include the fitting and installing of the above.

The time allowed for completion was 36 months, or to February 15th, 1904. For various causes, this time was extended to May 12th, 1906.

The main engines are to develop 19,000 indicated horsepower when making 120 revolutions per minute.

The guaranteed speed of the ship is 19 knots per hour for four hours. For failure to attain this speed a penalty was to be inflicted at the rate of $50,000 per quarter knot between 18½ and 19 knots, and $100,000 per quarter knot between 18 and 18½ knots. If the speed fell below 18 knots the Government has the option of accepting the vessel at a reduced price.

The complement is 772 men and 49 officers.

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet	435
on L.W.L., feet	435
over all, feet and inches	441-01
Beam, extreme, feet and inches	76-02¼
at load-water line, feet and inches	76-02¼
Depth molded, main deck, feet and inches	41-06
Draught, mean normal, feet and inches	23-09
on trial, feet and inches	23-09
Displacement at trial draught, tons	14,680
normal draught, tons	14,680
per inch at L.W.L., normal draught, tons	61

Area of immersed midship section, square feet	1,737
L.W.L. plane, normal draught, square feet	25,610
Center of gravity of L.W.L. plane aft of midship section, feet	2.61
buoyancy above base line, feet	12.80
aft of midship section, foot	.19
Transverse metacenter above center of buoyancy, feet	19.13
Longitudinal metacenter above center of buoyancy, feet	515
Coefficient of fineness, block	.663
midship section	.96
L.W.L.	.772
Cylindrical coefficients, longitudinal	.690
vertical	.858
Number of frames	108

The hull is built of basic open-hearth steel, with frames spaced 4 feet apart except under the machinery spaces. The inner bottom extends from frame 13 to frame 95, the double-bottom compartments between frames 60 and 69 are arranged for reserve feed-water tanks. A cofferdam 36 inches wide extends the whole length of the ship on the protective deck, extending 3 feet above the berth deck, packed with cellulose forward and aft of the midship armor. The superstructure which supports the bridge deck extends between the 12-inch turrets, but is not the full width of the ship. It is recessed amidships for the broadside 8-inch turrets, and is kept back from the side of the ship sufficiently to allow of a fore-and-aft train of these turrets. Over the forward end of the superstructure is the bridge, and above that the flying bridge. On the forward bridge are the chart house, emergency cabin, conning tower and torpedo-directing stations. Two 3-pdr., two 30-caliber automatic and two 30-caliber machine guns are on this bridge. On the flying bridge are the compasses, steering apparatus and searchlights. There is a bridge over the after end of the superstructure on which are two 3-pdr. and four 30-caliber automatic guns. On the superstructure are four 3-pdrs. forward and the same number aft. The following boats are carried on the bridge deck:

1 50-foot steam cutter;
1 36-foot steam cutter;
2 36-foot launches;

4 30-foot cutters;
2 20-foot dingheys;
1 16-foot dinghey.

They are handled by two 5-ton, electrically-operated boat-cranes forward, and one 15-ton and one 8½-ton crane aft.

The main deck extends from stem to stern, and is partially covered by the superstructure between frames 32 and 78. The two superposed turrets with two 12-inch and two 8-inch guns each, and the two broadside turrets with two 8-inch guns each, are on this deck.

Mounted in the superstructure are four 3-inch guns. The hawse pipes extend through the bow to this deck, and the anchor chains lead to the windlass house, which projects through this deck. Inside the superstructure, from forward aft, are the crew's galley, drying room, bakery, officers' galley, paymaster's office, executive officer's office, officers' bathrooms, officers' waterclosets and wireless telegraph station. The following boats are carried on davits on this deck:

2 30-foot cutters;
2 20-foot dingheys forward and two 30-foot whaleboats;
1 30-foot barge and one 30-foot gig aft.

On the gun deck are mounted twelve 6-inch guns, four 3-inch guns forward and four 3-inch guns aft. Forward are the crew's shower baths and lavatories, amidships the berthing spaces, and aft the quarters of the admiral, captain, chief of staff, executive, navigator, chief engineer, paymaster, and two wardrooms. On the berth deck are forward, the sick bay, operating room, isolation ward, dispensary and bath, chief machinist's and chief petty officers' quarters and lavatories, staterooms of the master-at-arms and sergeant of marines, laundry, refrigerating plant and brig. Amidships is the crew space and aft the officers' quarters. The protective deck, which extends the full length of the ship, is 3 inches thick on the slope and 1½ inches on the flat. On the upper and lower platforms and in the hold are the magazines, storerooms, torpedo and mine rooms, etc. The protection of the vessel consists of an armor belt at the water line 8 feet in width, and a maximum

thickness opposite the machinery spaces of 11 inches from the top down 5 feet, where it tapers to 8 inches at the bottom. Forward and aft of this belt the armor decreases in thickness by steps until at the bow and stern it has a uniform thickness of 4 inches. Above this main belt and covering the central portion of the ship for a distance of 245 feet is armor of a uniform thickness of 6 inches extending up to the main deck and connected to the aft barbette by athwartship armor 6 inches thick, and joined forward by inclined armor 6 inches thick, thus forming a central casemate.

The 12-inch barbettes, which are supported on the protective deck, are 10 inches thick in front and $7\frac{1}{2}$ inches thick in back. The 8-inch barbettes, which rest on the main deck, are 6 inches thick in front and 4 inches in back. The subbarbette, from the main to the gun deck, is 4 inches thick. From the gun deck an armored tube protecting the ammunition hoists extends through the protective deck.

The 12-inch guns are protected by armor 12 inches thick on the slope, 8 inches thick on the side and 6 inches thick at the back. The armor for the 8-inch superposed guns is 12-inches thick on the slope, 8 inches on the sides and 6 inches in back. The armor for the 8-inch broadside guns is $6\frac{1}{2}$ inches thick on the slope, 6 inches thick on the sides and back.

ELECTRIC PLANT.

The generating plant consists of two dynamos of 100 kilowatt and six of 50 kilowatt capacity at 125 volts. They are in four rooms, two at the level of the hold and two above, with the center-line bulkhead separating each pair.

The generators are of General Electric make, while the engines are from B. F. Sturtevant Co.

The lighting outfit comprises in the various circuit fixtures for 1,154 lights and 6 searchlights.

The power outfit comprises 112 motors, as follows:

4 turret-turning motors of 35 H.P. each;

4 turret-turning motors of 15 H.P. each;

4 ammunition-hoist motors of 30 H.P. each;

8 ammunition-hoist motors of 8 H.P. each;
21 ammunition-hoist motors of 2½ H.P. each;
5 whip-hoist motors of 3 H.P. each;
4 gun-elevating motors of 5 H.P. each;
8 gun-elevating motors of 3½ H.P. each;
4 rammer motors of 5 H.P. each;
8 rammer motors of 5 H.P. each;
5 boat-crane motors of 35 H.P. each;
3 boat-crane motors of 20 H.P. each;
1 workshop motor of 10 H.P.

33 ventilation-fan motors varying in sizes from ½ H.P. to 15 H.P. each.

DYNAMO ENGINES.

Type	Compound.
100-kw. dynamos, number	2
Diameter of cylinder, H.P., inches	10
L.P., inches	18
Stroke, inches	10
50-kw. dynamos, number	6
Diameter of cylinder, H.P., inches	08
L.P., inches	14
Stroke, inches	08

PROPELLING MACHINERY.

There are two four-cylinder triple-expansion outboard-turning engines of the vertical, inverted, direct-acting type, in two watertight compartments, separated by a fore-and-aft watertight compartment. The order of the cylinders from forward is F.L.P., H.P., M.P., A.L.P. The H.P. and M.P. cranks are at 90 degrees, as are also the two L.P. cranks. The H.P. and F.L.P. cranks are opposite. The sequence of the cranks is H.P., M.P., F.L.P., A.L.P. The cylinders are supported by twelve forged-steel columns, which are tied fore and aft by heavy cast-steel I beams on the back supporting the crosshead guides, and on the front by a forged-steel tie. Additional forged-steel diagonal ties are fitted on both sides. Each pair of columns is tied together athwartship by diagonal braces. The valves are of the single-ported piston type, with cast-iron heads and packing rings and steel followers. The

H.P. valve takes steam on the inside, the others on the outside. The H.P. cylinder has one valve, the others two each. All valves are arranged to draw from the top. Each valve has a balance piston working in a cylinder cast in the cover.

The valve gear is Stephenson double-bar link type, and the H.P. valve stem connects with the link direct, while the M.P. and L.P. valve stems are connected by crossheads.

The engine is controlled by a floating-lever reversing engine with oil-cylinder control, and is connected to a reverse shaft supported from the engine columns. The reverse shaft carries arms in which independent adjustable cutoff blocks are fitted to connect with each link, giving a range in cutoff from .5 to .74 of the stroke. All the cylinders are jacketed around the working liners and at both ends, except the H.P., which has no jacket on the ends.

Each engine has a throttle valve balanced by a piston working in a cylinder cast with the valve casing, and operated by a hand wheel at the working platform. The H.P. piston is of cast iron, the others of cast steel, all having forged-steel followers. Each piston has two packing rings, each in one piece cut diagonally, with tongue pieces over the joint. The rings are put in the pistons so that the joints come opposite each other. The piston rods are hollow, of class A nickel-steel, oil tempered and annealed. They are ground cylindrical and polished and tapered at each end to fit the piston and the crossheads. The crossheads are of nickel-steel forged with the crosshead pins and are bolted to cast-steel slippers. The go-ahead guides are of cast iron, hollow, to provide for water circulation, and are supported at the top by lugs on the cylinders and at the bottom by the cast-steel tie rods. The backing guides of cast iron are bolted to flanges on the go-ahead guides. The connecting rods are of the same material as the piston rods, are hollow and forked at the upper end to carry the crosshead brasses. The crank-pin boxes are of composition and are held to the stub end of the connecting rod by a steel cap and through bolts. The eccentrics are in two parts, the larger of cast iron and the smaller of wrought steel. The

eccentric straps are of composition and the rods of nickel-steel.

The turning engine is a double, inverted engine located at the aft end of the bedplate and connected to the crankshaft by worm gearing. The working platforms are inboard, where are located the throttle-valve wheel, reversing and drain levers, clock, gauges, counters, etc.

ENGINE DATA.

Cylinders, number for each engine	4
H.P., diameter, inches	35
M.P., diameter, inches	57
L.P., diameter, inches	66
Stroke of all pistons, inches	48
Valves, H.P., one for each cylinder, inches	20
M.P., two for each cylinder, inches	22
L.P., two for each cylinder, inches	24
Valve stems, H.P., diameter, inches	03
M.P., diameter, inches	03
L.P., diameter, inches	03
diameter through valve, H.P. and I.P., inches	01 11/16
L.P., inches	02 1/4
at balance piston, H.P. and I.P., inches	01 1/2
L.P., inches	01 3/4
Main steam pipe, diameter at throttle, inches	12
Area of cross section, square inches	113
Net area of both M.P. to H.P. piston	2.65
L.P. to H.P. pistons	7.11
Piston-rod diameter, inches	08
of axial holes, H.P., inches	01 1/4
M.P., inches	03 3/4
L.P., inches	05
length, feet and inches	9-04 5/16
Cylinder wall, H.P., thickness, inches	02
M.P., thickness, inches	01 1/4
L.P., thickness, inches	01 1/4
liners, H.P., thickness, inches	01 1/4
M.P., thickness, inches	01 3/8
L.P., thickness, inches	01 1/4
Valve-chest liners, thickness, inches	01 1/4
Jacket space, inch	00 3/4
Cylinder relief valves, two to each cylinder, inches	03 1/2
Connecting rod, length center to center, inches	96
diameter upper end, inches	07 1/4
lower end, inches	09
axial hole, H.P., inches	01 1/4

Connecting rod, diameter axial hole, M.P., inches	04½
L.P., inches	05
crosshead bolts (four), inches	3½×26½
crank-pin bolts (two), inches	4½×37½
Crosshead surface, each, square inches	690
backing, each, square inches	525
pin, diameter, inches	09½
diameter of axial hole, H.P., straight, inches	01½
I.P. and L.P., double taper, inches	2 to 6
length of pin, inches	10 11/16
Crankshaft, number of sections	2
diameter, inches	17½
of axial hole, inches	10½
coupling discs, inches	32
thickness coupling dics, inches	03½
coupling bolts in one flange (eight), diameter, inches	03½
journals, diameter, inches	17½
length forward to aft, inches	21½, 21½, 21½, 21½, 29, 21½
length of forward section, feet and inches	16–06½
after section, feet and inches	17–10
Crank pin, diameter, inches	18
length, inches	21½
axial hole, inches	10½
webs, width, inches	20
thickness, inches	11½
Thrust shaft, diameter, inches	16½
axial hole, inches	10½
Thrust shaft, collars, number, on each shaft	13
diameter, inches	26
thickness, inches	02
space between, inches	04
total thrust surface each engine, sq. inches	3,952
length, feet and inches	14–09
steady bearing, diameter, inches	16½
length, inches	12
Line shaft, diameter, inches	16½
axial hole, inches	10½
coupling disc, inches	32
coupling bolts in one flange	8
diameter, inches	03½
length, feet and inches	29–00
Propeller shaft, without casing, diameter, inches	17½
in bearings, inches	19½
length, feet and inches	26–03
Stern-tube shaft, after bearing, diameter, inches	19½
length, feet	6
forward bearing, diameter, inches	19½
length, feet and inches	5–07

Inboard coupling, diameter, inches	32
length, inches	19¼
Outboard coupling, diameter, inches	23
length, feet and inches	6–00
Stern-bracket bearing, diameter, inches	19½
length, feet and inches	7–02
Reversing engine, diameter steam cylinder, inches	15
oil cylinder, inches	07½
stroke, inches	21½
Turning engine, diameter cylinders (two), inches	06
stroke, inches	06
Clearance of H.P. cylinder, per cent.....top, 24.7; bottom,	23.7
M.P. cylinder, per cent.....top, 21.0; bottom,	20.7
F.L.P. cylinder, per cent.....top, 19.4; bottom,	29.2
A.L.P. cylinder, per cent.....top, 19.3; bottom,	19.4

MAIN CONDENSERS.

There is one main condenser in each engine room. The shell is of steel, the water heads of composition, the tubes of Admiralty metal not tinned. There is a by-passed valve in the division plate of the water chest at the injection and delivery end to allow the circulating pump to discharge directly overboard.

Diameter of shell, inside, feet and inches	6–10
Thickness of shell, inch	00⅝
Length over heads, feet and inches	15–08
Tubes, diameter, outside, inch	00¾
length between tube sheets, feet and inches	13–00
thickness, inch	01
spacing between centers, inch	00 13/16
number in each condenser	5,583
cooling surface, each, square feet	11,875

MAIN AIR PUMPS.

Forward of the condenser in each engine room there is a two-cylinder, vertical, double-acting air pump of the Fore River design.

Diameter of steam cylinders, each, inches	14
water cylinders, each, inches	31½
Stroke, inches	21
Diameter of piston rods, inches	02½
pump rods, inches	04½
pump valves in bucket, inches	05¾
Number of delivery valves, 30; diameter of each, inches	05¾
Area through valve openings, square inches	16

MAIN CIRCULATING PUMPS.

There is one centrifugal, double-inlet, circulating pump driven by compound, vertical, inverted engine, located aft of the main condenser. Each pump has a capacity of 12,000 gallons per minute, at 266 revolutions per minute, and it may draw from the sea, main drain or the engine-room bilge. The three suctions are so interlocked that only one can be open at a time.

Diameter of H.P. cylinder, inches	10
L.P. cylinder, inches	20
Stroke, inches	10
Diameter of pump runner, inches	42
Width of pump runner at periphery, inches	05
hub, inches	12
Diameter of inlet nozzle, inches	20
outlet nozzle, inches	19½

FEED AND FILTER TANKS.

Under each condenser is a combined feed and filter tank of 4,200 gallons capacity. A horizontal partition divides each tank into two compartments, of which the upper is the filter tank, of 1,150 gallons capacity, and the lower the feed tank proper, of 3,050 gallons capacity. Each filter tank is divided into five parts by vertical partitions, and the water flows through the filtering material four times on its way to the tank.

FEED PUMPS.

There are two main feed pumps of Fore River design in each engine room. They draw from the hotwell-pump discharge and discharge into the main feed lines.

In each of the six firerooms is one auxiliary feed pump of Fore River design bolted to the center-line bulkhead. Each pump may draw from the feed tanks, from the hotwell-pump discharge, from the main drain, the auxiliary drain, the sea, the bilge in its own compartment and from the bottom blows of all boilers in its compartment. It may discharge into the auxiliary feed line, the fire main or overboard.

Main feed pumps, double-acting, single, vertical, piston type.

Diameter of steam cylinder, inches..... 18
water cylinder, inches..... 10
Stroke, inches..... 18

Auxiliary feed pumps, double-acting, single, vertical, piston type.

Diameter of steam cylinders, inches..... 14
water cylinders, inches..... 08
Stroke, inches..... 12

FEED-WATER HEATERS.

In each engine room there is one feed-water heater containing 950 square feet of heating surface in 666 tubes. Steam from the auxiliary exhaust is by-passed into the heater and the back pressure is regulated by a spring-relief valve. The hotwell pump delivers water through this heater to the feed-pump suction.

HOTWELL PUMP.

There is one vertical, double-acting, single-piston, hotwell pump in each engine room, which may draw from either air-pump suction, from hose connections on the side of the ship, and from the main and reserve-feed tanks. It may discharge into the feed-pump suction either directly or through the feed-water heater, into the auxiliary feed line or into the main or reserve-feed tanks.

Diameter steam cylinder, inches..... 12
water cylinder, inches..... 14
Stroke, inches..... 12

FIRE AND BILGE PUMPS.

There is one vertical, double-acting, single-piston, packed fire and bilge pump in each engine room. These pumps draw from the sea, the drainage system and from the bilge. They discharge into the fire main or overboard.

Diameter of steam cylinder, inches..... 12
water cylinder, inches..... 10
Stroke, inches..... 12

AUXILIARY CONDENSERS.

In each engine room there is one auxiliary condenser having 602 square feet of cooling surface, with combined air and circulating pump attached.

Diameter of steam cylinder, inches	08
water cylinder, inches	10
air cylinder, inches	10
Stroke, inches	12

In the dynamo room there is one auxiliary condenser having 1,207 square feet of cooling surface, with a combined air and circulating pump attached, of the following dimensions:

Diameter of steam cylinder, inches	12
water cylinder, inches	14
air cylinder, inches	14
Stroke, inches	12

FORCED-DRAFT BLOWERS.

The closed-fireroom system of forced draft is used, there being two Sturtevant blowers for each fireroom.

Number of blowers	12
Type of engine	Two-cylinder, vertical.
Diameter of steam cylinders, inches	06
Stroke, inches	05
Diameter of piston rods, inches	$01\frac{5}{16}$
fan, inches	66

MACHINE SHOP.

Located on the upper platform between the aft fireroom and the engine room and extending athwartship to the width of the engine room is the general workshop. This is divided into two parts by a fore-and-aft watertight bulkhead. Communication between the two is by means of a watertight door. The tools are driven by a 10-horsepower motor. The following tools are installed:

1 30-inch and 59-inch swing back-geared gap lathe, 10 feet between centers;

1 14-inch screw-cutting back-geared engine lathe, 4 feet between centers;

1 16-inch back-geared crank shaper;
1 36-inch Dreses simplex radial drill, to drill up to 1½ inches;
1 16-inch sensitive drill;
1 No. 1 Universal milling machine;
1 combined hand punch and shears;
1 double emery grinder;
1 30-inch grindstone;
7 bench vises.

BLACKSMITH SHOP.

Located on upper deck amidships, one motor-driven forge and the usual equipment of tools is installed.

ASH HOISTS.

There are six ash hoists, made by Williamson Brothers. They are located in the fireroom hatches, and the buckets are hoisted through the ventilators to the gun deck and trollied to the ash chutes at the sides.

EVAPORATING AND DISTILLING PLANT.

There are four evaporators of the Williamson Brothers make, each having a capacity of 4,750 gallons in 24 hours and having a heating surface of 240 square feet each. There are six distillers of a combined capacity of 20,000 gallons of water in 24 hours, each having a cooling surface of 66 square feet.

The evaporators are located on the berth deck at the after end of the boiler hatch, and the distillers are located in the hatch one deck higher.

ICE MACHINE.

One Allen dense-air ice machine of three tons daily capacity is installed forward on the port side of the berth deck, with the ice-making boxes near. Cooling pipes lead to the four refrigerating rooms and to the scuttle butts.

SCREW PROPELLERS.

The propellers are of manganese-bronze, and turn outboard when going ahead, the starboard propeller being right-hand. They are true screws with the surfaces ground smooth and tinned.

Number of blades	3
Diameter, feet and inches	17-09
of hub, feet and inches	4-08
Length of hub, feet and inches	3-07
Pitch, feet and inches	18-08
Helicoidal area, square feet	99.2
Projected area, square feet	82.5
Pitch ratio	1.06
Area ratio	.4

BOILERS.

There are twelve Babcock & Wilcox boilers of the latest improved type, arranged in six watertight compartments.

Floor space, length, feet and inches	9-01½
width, feet and inches	17-09½
Length of grate, feet and inches	7-00
Width of grate, feet and inches	16-00
Grate surface one boiler, square feet	111.8
all boilers square feet	1,342
Heating surface, one boiler, square feet	4,682
all boilers, square feet	56,184
Ratio H.S. to G.S	41.87
Number of tubes, one boiler, 2-inch	946
4-inch	62
total	1,008
elements one boiler	30
Weight of water in one boiler at steaming level, pounds	16,454
one boiler complete with fittings, pounds	96,545

HEATING SYSTEM.

Radiators and heaters throughout the ship are arranged in circuits, each circuit being so connected that it can be operated independently of the others. The circuits are in turn arranged in groups, each group having a receiving manifold and a drainage manifold. Steam is taken from the auxiliary steam

pipe at the receiving manifold. Here there is a stop valve for each circuit and a stop valve, an ajustable reducing valve, and a steam gauge at the connection with the auxiliary steam pipe. The drainage manifolds have a stop valve and a check valve for each circuit. Each manifold discharges through a trap with a by-pass to the feed tanks, the auxiliary condenser or the bilge. By this arrangement each circuit can be entirely isolated and water from other circuits prevented from backing up into it.

OFFICIAL TRIALS.

STANDARDIZATION RUNS.

For the purpose of standardizing the screws runs were made over the measured-mile course off Rockland, Maine, March 28, 1906. The weather was overcast to clear and cold, with moderate to gentle breezes. The sea was smooth.

Just previous to the vessel starting on the standardization runs the draught was—

Forward, feet and inches,	23–06 $\frac{5}{16}$
Aft, feet and inches,	24–00 $\frac{9}{16}$
Mean, feet and inches,	23–09 $\frac{7}{16}$
Corresponding displacement, tons,	14,910

These runs gave the data recorded in Tables I and II, from which the curves on Plate I were drawn.

After the standardization runs the draught was—

Forward, feet and inches,	23–04
Aft, feet and inches,	23–11 $\frac{1}{4}$
Mean, feet and inches,	23–07 $\frac{5}{8}$
Corresponding displacement, tons,	14,810

OFFICIAL FOUR-HOURS' TRIAL.

On the morning of March 29, 1906, the vessel got underway for the official four-hours' trial.

Table I.—U. S. S. *NEW JERSEY.*

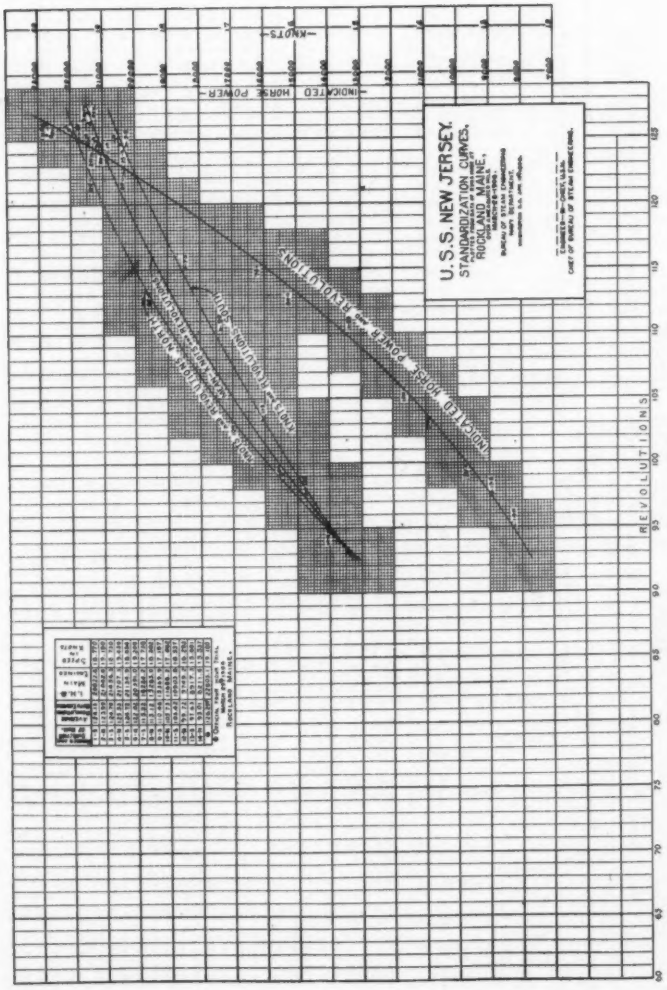
Data from Progressive Runs over Measured-Mile Course off Rockland, Me., March 28, 1906.

Number and direction of run.	Time on course.	Revolutions of main engines.					Speed in knots per hour.	Indicated horsepower.											
		To make 1 knot.		Per minute.				Starboard engine.					Port engine.					Total of both main engines.	Auxiliaries in operation.	Total for all machinery.
		Starboard.	Port.	Starboard.	Port.	Mean.		H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.	H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.			
	H. M. S.																			
1, S.	3-11.8	396.5	397.0	124.0	124.2	124.1	18.77	3,296.9	3,343.2	2,246.9	1,818.0	10,705.0	3,061.3	3,477.0	1,646.5	1,932.8	10,117.6	20,822.6	669.8	21,492 4
2, N.	3-07.7	385.9	388.8	123.5	124.4	124.0	19.18	3,224.9	3,518.1	2,267.8	1,830.5	10,841 .3	2,770.4	3,498.9	1,782.8	2,049.2	10,101.5	20,942.6	723.7	21,666.3
3, S.	3-12.2	398.5	401.7	124.2	125.2	124 8	18.73	3,245.5	3,728.9	2,086.7	2,199.1	11,260.2	2,659 9	3,628.4	2,010.6	1,897.2	10,196.1	21,456.3	700.0	22,156.3
4, N.	3-05.0	385.7	388.3	125.1	125.9	125.5	19 46	3,339.2	3,755.0	2,194.0	1,957.0	11,245.2	2,717.9	3,741.6	2,053.2	1,949 4	10,462.1	21,707.3	689.4	22,396.7
5, S.	3-13.2	401.8	401.3	124.8	124.6	124.7	18 63	2,980.0	3,898 5	2,363 2	1,983.0	11,164.7	2,902.2	3,527 4	1,898.6	1,642.0	9 970.2	21,134 9	683.6	21,818.5
6, N.	3-07.5	382.1	380 1	122.3	121.7	122.0	19.20	3,571.0	3,222.2	2,074.9	1,692.2	10,560 3	3,510.8	3,093 6	1,633.2	1,593.1	9,830.7	20,391.0	680.7	21,071.7
7, S.	3-22.95	390 0	389 4	115 3	115.1	115.2	17.74	2,803.0	2,542.9	1,728.0	1,281.7	8,355.6	2,837.2	2,510.3	1,251.1	1,270.0	7,868.6	16,224.2	518.4	16,742.6
8, N.	3-16.7	370.3	372.0	112.8	113.4	113.1	18.30	2,742.6	2,350.0	1,486.4	1,226.3	7,805.3	2,556.2	2,389.4	1,185.7	1,148.4	7,279 7	15,085 0	483.1	15,568.1
9, S.	3-29.45	384.6	386.1	110.2	110.7	110.5	17.19	2,504.0	2,160.6	1,007.4	1,052.7	6,724.7	2,476.9	2,190.6	1,020.8	956.9	6,645.2	13,369 9	405.5	13,775 4
10, N.	3-30 75	369.9	373 2	105.4	106.1	105.7	17 08	2,190.3	1,839.6	832.9	876.3	5,739 1	2,228.9	1,929 8	908.3	882.1	5,949 1	11,688.2	402.3	12,090.5
11, S.	3-37.7	374.5	375.4	103.3	103.5	103.4	16.54	2,111 9	1,708.4	739.9	807.9	5,368.1	2,093.4	1,795.0	809.9	837.4	5,535.7	10,903.8	349.6	11,253.4
12, N.	3-40.95	370.3	364.1	100.5	99.9	100.2	16.29	2,010.1	1,601.5	645.7	761.7	5,019.0	1,819.5	1,502.2	683 9	724.6	4,730.2	9,749 2	355.2	10,104.4
13, S.	3-46.55	365.9	370.4	97.0	98.2	97.6	15.89	1,708.2	1,396.7	535.2	655.1	4,295.2	1,807.4	1,456.1	679.3	679.3	4,622.1	8,917.3	306.6	9,223.9
14, N.	3-51.7	366.6	367.1	94 9	95.1	95.0	15.54	1,660.7	1,308.5	508.1	594.1	4,071.4	1,641.8	1,292.6	587.1	618.4	4,139.9	8,211.3	289 7	8,501.0

Table II.—U. S. S. *NEW JERSEY.*

Data from Progressive Runs over Measured-Mile Course, off Rockland, Me., March 28, 1906.

Number and direction of run.	Which engine.	Steam pressures.			Mean effective pressures.				Vacuum, in inches.	Air pressure, in inches of water.	Cut-off of cylinders from beginning.				Temperatures.						Revolutions or double strokes.						
		In boilers above atmosphere.	At engine above atmosphere.	H.P. steam chest, absolute.	In 1st receiver, absolute.	In 2d receiver, absolute.	H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Referred to L.P.		H.P.	I.P.	Forward L.P.	Aft L.P.	Outside air.	Engine-room working platform.	Fireroom working level	Injection.	Discharge.	Hotwell.	Feed.	Main feed pump.	Hotwell pump.	Air pump.	Dynamo condenser pump	Sanitary pump.	Circulating pump.	Dynamo amperes.	Fireroom blowers.	Pyrometer, degrees, 49 feet above grate.
1, S.	S.	269.8	255	255	260	105	35	117.0	44.0	22.0	17.8	52.41	26.5	1.57	Full.					46	64	65	34	99	85	148	84.5	89	13	89	24	130	460	476.6	538
	P.		255	260	105	37	108.5	45.7	16.1	18.9	49.47	26.5	26.5								64			93	73	150	85.5	39	19			200		469.0	
2, N.	S.	272.8	265	265	265	97	37	118.5	46.5	22.3	18.0	53.80	26.5	1.63						46	62	65	34	98	72	142	84.5	36	18	89	27	180	460	478.6	562
	P.		260	265	265	115	39	98.0	45.9	17.4	20.0	49.96	26.0								64			94	72	141	84.5	36	11	200		460	469.0	
3, S.	S.	270.0	260	260	265	115	38	115.0	49.0	20.4	21.5	55.04	27.0	1.60						45	64	65	34	108	78	144	84.5	31	19	89	28	164	470	473.3	555
	P.		260	260	265	97	36	93.5	47.3	19.5	18.4	49.80	26.0								64			94	71	152	87.0	39	9	208		468	468.5	
4, N.	S.	276.0	265	265	265	97	37	117.5	49.0	21.3	19.0	54.59	27.0	1.65						47	62	66	34	104	76	144	84.5	33	19	89	26	160	470	462.5	570
	P.		265	266	120	38	95.0	48.5	19.8	18.8	50.44	26.0									64			93	72	148	85.0	38	10	210		470	462.5	
5, S.	S.	268.0	266	260	100	38	103.0	51.0	23.0	19.3	54.33	26.0	1.7							43	63	67	34	94	72	148	84.0	32	30	89	26	154	480	466.7	584
	P.		255	263	110	37	108.5	46.2	18.5	16.0	48.57	26.0									65			108	78	148	86.5	38	10	210		480	466.7	
6, N.	S.	269.3	260	260	260	86	32	128.5	43.0	20.6	16.8	52.43	27.0	1.65						41	63	65	34	104	78	144	84.5	34	21	89	27	162	480	462.5	561
	P.		260	265	90	31	127.0	41.5	16.3	15.9	49.05	26.5									62			91	72	148	87.0	24	10	212		480	466.7	
7, S.	S.	284.4	278	218	215	70	27	107.0	36.0	18.2	13.5	44.00	27.0	0.98						38	62	67	34	100	70	140	80.0	23	16	89	21	148	470	420.0	563
	P.		280	280	280	75	27	108.5	35.6	13.2	13.4	41.51	27.0								63			89	72	158	85.5	24	8	188		470	420.0	
8, N.	S.	279.1	273	220	220	65	27	107.0	34.0	16.0	13.2	42.01	27.0	1.1						40	64	65	34	98	72	158	85.5	18	16	89	28	152	470	411.0	517
	P.		273	220	75	25	107.0	34.4	12.7	12.3	40.06	26.0									64			91	66	156	81.0	23	13	132		470	420.0	
9, S.	S.	197.2	190	195	67	23	98.5	32.3	11.2	10.5	36.44	27.0	1.1							39	64	66	34	98	66	140	76.5	22	14	89	21	126	460	400.0	492
	P.		200	188	55	21	91.5	28.5	9.6	10.1	33.07	27.0									64			92	76	150	75.5	19	8	124		460	381.7	
10, N.	S.	198.5	195	190	60	21	92.5	29.7	10.4	10.1	34.06	26.5	0.97							42	64	65	34	92	68	148	78.0	17	15	89	19	136	460	400.0	475
	P.		195	190	60	21	90.0	27.0	10.4	9.5	31.56	27.5	0.82							41	70	65	34	90	74	136	73.5	25	18	95	31	112	475	353.2	465
11, S.	S.	211.0	272	185	55	20	89.0	28.3	9.5	9.8	32.46	27.0									74			92	63	154	81.0	15	19	95	18	128	460	350.0	462
	P.		215	170	48	18	88.0	26.0	9.2	9.2	30.31	27.5	0.70							41	76	65	34	89	70	178	82.0	14	19	95	30	132	475	353.2	462
12, N.	S.	213.8	210	170	50	18	81.0	24.8	8.4	8.9	29.04	27.0									76			88	64	139	75.0	17	18	95	30	132	475	353.2	462
	P.		170	180	55	43	77.5	23.5	6.7	8.2	26.88	28.0									74			90	70	148	73.5	18	6	85	30	122	478	333.3	451
13, S.	S.	190.7	188	165	49	17	81.0	24.2	8.4	8.4	28.57	27.0	0 50							42	77	65	34	85	64	138	72.0	16	18	85	30	120	478	287.5	451
	P.		215	150	40	16	77.0	22.5	6.5	7.6	26.04	26.5									74			83	70	154	76.0	19	11	85	30	122	478	333.3	446
14, N.	P.	219.6	220	157	43	16	76.0	22.2	7.5	7.9	26.44	27.0	0.57							42	78	65	34	85	64	139	4.5	16	10	92	28	122	478	287.5	


U.S.S. NEW JERSEY.
STANDARDIZATION CURVES.
ROCKLAND, MAINE,
INDICATED HORSE POWER
KNOTS
REVOLUTIONS

The draught was as follows:

Forward, feet and inches,	23–06$\frac{15}{16}$
Aft, feet and inches,	24–02$\frac{7}{16}$
Mean, feet and inches,	23–00$\frac{10}{16}$
Corresponding displacement, tons,	14,990

The weather was clear and pleasant, with light airs and breezes. The sea was smooth.

The average revolutions during the trial, per minute, were 126.297, corresponding to a speed of 19.18 knots.

A synopsis of the data obtained follows:

PERFORMANCE.—FOUR HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Steam pressure at boilers (per gauge), pounds	272.0	
H.P. steam chest (gauge), pounds	244.2	246.2
1st receiver (absolute), pounds	105.3	116.4
2d receiver (absolute), pounds	41.6	40.7
Vacuum in condensers, inches of mercury	26.83	26.80

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	39.6	40.1
Discharge, degrees	103.2	98.7
Hotwell, degrees	80.2	78.4
Feed water, degrees	147.0	156.2
Engine room, upper platform, degrees	80.7	78.0
working platform, degrees	67.9	71.4
Firerooms, above grating, degrees	66.8	66.8
working level, degrees	64.7	64.7
Base of smoke stacks, degrees	589.4	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Main engines (mean for trial)	126.27	126.32
average for both	126.30	
Pumps, main, air	12.6	11.8
circulating	202.2	190.5
feed	25.4–25.5	30.1–24.3
hotwell	34.7	37.3
bilge	22.2	8.2
dynamo condenser	6.0	

	Starboard.	*Port.*
Blower engines	484.0	494.0
Speed of ship, in knots per hour	19.18	
Slip of propeller, in per cent. of its own speed, based on mean pitch and average speed	17.25	17.16
Air pressure in firerooms, inches of water	1.84	1.84

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Averages of cards taken in half-hourly periods.*)

Main engines, H.P. cylinder	101.7	89.06
I.P. cylinder	54.2	56.27
F.L.P. cylinder	19.9	21.04
A.L.P. cylinder	21.4	22.2
Mean equivalent pressure, in pounds per square inch referred to combined area of L.P. pistons	54.82	54.83

INDICATED HORSEPOWER.

Main engines, H.P. cylinder	2,916.3	2,554.9
I.P. cylinder	4,191.3	4,354.4
F.L.P. cylinder	2,066.5	2,188.3
A.L.P. cylinder	2,223.8	2,309.6
total	11,397.9	11,407.2
both engines	**22,805.1**	

AUXILIARIES.

Air pumps	4.6	4.35
Circulating pumps, main	72.0	61.5
Feed pumps, main	21.0–21.2	27.5–19.8
Hotwell pumps	25.0	27.0
Bilge pumps	5.2	0.5
Sanitary pump	5.7	
Dynamo-condenser pump	0.3	
Blower engines (forced draft)	394.7	
Dynamo engine (one in operation)	75.0	
Total	**765.35**	
all machinery in operation	**23,570.4**	

COAL.

Kind and quality	Pocahontas, screened, good.
Pounds, per hour	52,360.0

Deduced Data.

I.H.P. (total) per square foot of grate surface	17.56
heating surface	.419
Pounds of coal per I.H.P. per hour (main engines only)	2.29
all machinery in operation	2.22
Pounds of coal per square foot of grate surface	39.02
heating surface	.932
Cooling surface (main condenser), square feet per (total) I.H.P.	1.008
Heating surface, square feet per (total) I.H.P.	2.384

AFTER THE TRIALS.

The working of the machinery, both main and auxiliary, and its performance during the trial, were entirely satisfactory. The boilers steamed finely and worked in a satisfactory manner throughout.

A careful inspection of the various parts of the machinery during and after the trial showed the engines, boilers and appurtenances to be in excellent condition.

CHARACTERISTICS IN DESIGN AND ARRANGEMENT OF MARINE TURBINES AND PROPELLERS.

By Ernest N. Janson, Member.

I.—PARSONS SYSTEM.

Cause and effect of expansion and flow of steam through a turbine is now generally well understood, and the operation may be applied in analysis for purpose of designing marine turbines in a manner similar to that governing stationary turbine design, for which a large amount of the fundamental data are elaborately discussed in various scientific works and, in a general way, have also been given in professional papers and tests. Much desirable data, however, is wanting, with respect to results obtained with some of the later marine installations. That this information is withheld is to be regretted, as its possession would be of great value, commercially as well as strategically, for comparison with present systems as well as for reference in future installations.

Generally speaking the steam action is alike in both the marine and the stationary turbine, but with respect to details and arrangement the two types differ materially. Among the principal differences between them, we note for the former—

1. That the total range of expansion is divided among three or more independent turbines, each of which is doing such part of the total work as corresponds to the initial and final pressures of the steam entering and leaving each turbine.

2. Provisions for reversing the motion by separate backing turbines.

3. That the unbalanced pressure endwise in the vane system and on the turbine rotor is largely counterbalanced by the thrust of the screw, and, therefore, the elimination of balance pistons within the turbine.

4. An arrangement of separate low-speed and low-power turbines to answer cruising conditions when used in naval vessels.

5. Lower vane and steam speeds with an increased number of "expansions" and of buckets. Reduced stresses owing to smaller centrifugal forces, and, with dynamic disturbances greatly diminished, a more easily procured running balance.

Dimensions of turbines are influenced directly by—

Speed of revolutions, horsepower, initial steam pressure and the pressure of exhaust.

Brake Horsepower.—The indicated horsepower needed for propulsion is fixed with considerations and reference to the ship, means of power transmission and the propeller, and is obtained directly from the general equation:

$$\text{Propulsive efficiency } Pc = \frac{\text{Effective horsepower}}{\text{Indicated horespower}} = \frac{E.H.P.}{I.H.P.}$$

should Pc and E.H.P. be known.

There being no satisfactory means of getting the indicated horsepower of a steam turbine, the determination of its brake horsepower becomes imperative. This can be done by actual shop tests, using, for instance, the hydraulic disc brake, with which up to 10,000 or more brake horsepower may be measured. The propulsive efficiency constant will appear relatively larger when using brake horsepower instead of the indicated, and its relative value may be found by dividing Pc by the mechanical efficiency assumed (unless known) to exist in an engine developing sufficient power to give E.H.P. in the foregoing expression and substituting B.H.P. for I.H.P. In carefully designed and constructed marine engines the total loss of power from dead-load friction and the friction of the working load should not exceed ten per cent., and is actually not more than six per cent. in large installations, as proved by late tests, and a value lying between those limits may be used for engine efficiency.

To get the effective horsepower necessitates a predetermination of the effective thrust (T) required to overcome frictional and wave-making resistances of the hull at a stipulated

speed (K). This thrust is ascertained either by model experiments or by some method of comparison with known hulls. In a few instances it has been determined by towing the actual ship, when the pull in the tow rope, as measured by a dynamometer, will indicate said resistances.

The brake horsepower at the turbine end of the shaft must be large enough to overcome not only foregoing resistances but also frictional resistances of line shafting, frictional and other resistances of the propeller, together with the reaction on the hull caused by "thrust deduction" due to suction of the propeller when working in close proximity to the ship. A slight gain in the propulsive horsepower is obtained by the velocity of the water influenced by the wake. Then, with the following designations,

Brake horsepower of turbine, ... $= B.H.P.$

Propeller efficiency, ... $= Ep.$

Line shaft efficiency, ... $= Es.$

Thrust deduction $\left\{\begin{array}{l} t \text{ being equal to the thrust} \\ \text{required over that necessary} \\ \text{to exert previously mentioned} \\ T, \text{ expressed by a certain per-} \\ \text{centage of this } T \end{array}\right\} = 1 + t.$

Wake gain (w is the speed of wake in per cent. of speed of ship) $= 1 - w$.

Speed of ship, in knots, per hour $= K$.

Effective thrust in pounds (or tow-rope resistance) $= T$, we get

$$B.H.P. \times \frac{Ep \times Es}{(1+t)(1-w)} = E.H.P. = \frac{T \times 101.3\ K}{33{,}000} = \frac{T \times K}{325.8};$$

and if

$$\frac{Ep \times Es}{(1+t)(1-w)} = \text{propulsive efficiency} = Pc,$$

we get

$$B.H.P. = \frac{E.H.P.}{Pc}.$$

Scrutiny of the above formulae develops a state of uncertainty as regards the values to be assigned to the quantities entering.

The mechanical efficiency of the line shafting varies with number of bearings, diameter of shafting and the rigidity of supports, and thus is not constant. The propeller efficiency is not known actually, but only experimentally in comparison with models. The loss from "thrust deduction" $(1+t)$ and the gain from "wake" $(1-w)$ may only be surmised for the varying conditions under which propulsion is performed. These factors are often assumed to neutralize one another, under which assumption Pc becomes the product of Ep and Es. In the majority of cases, however, an allowance must be made for additional power to overcome "thrust deduction," by giving to "t" a value such that $(1+t)$ $(1-w)$ becomes larger than unity.

This product in all probability is seldom less than 1.02 and very often as high 1.05, and then $Pc = \frac{Ep \times Es}{1.02 \text{ to } 1.05}$.

In turbine installations we may assume Ep between .58 and .65 and Es between .97 and .99 when Pc, taking minimum efficiency in each case and making $(1+t)$ $(1-w) = 1.05$, becomes $= \frac{.97 \times .58}{1.05} = .536$ per cent. and $B.H.P. = \frac{E.H.P.}{.536} = \frac{T \times K}{325.8 \times Pc}$.

Example:

$T = 100{,}000$ pounds found, for instance, by the law of comparison applied to the actual hull and the model, for which the effective thrust has been ascertained by tank experiments.

$K = 24$ knots.

$Pc = .536$.

$$B.H.P. = \frac{100{,}000 \times 24}{325.8 \times .536} = 13{,}740.$$

Should the indicated horsepower be desired, then Pc must be multiplied by the engine efficiency, say .9, and will be $.536 \times .9 = .482$ when I.H.P. $= \frac{E.H.P.}{.482} = 15{,}460$.

Speed of Revolutions.—The rotative speed of a turbine must be determined with regard to circumstances, and, on the whole is governed by the speed most suitable to the apparatus receiving and transmitting the power, such as propeller, dynamo, pump or blower. In the marine turbine the propeller is the deciding element as to the upper limit of speed, while the lower limit is determined by consideration of weight, space and cost of the turbine.

Steam Pressure and Vacuum.—The usual steam pressure employed at entrance of H.P. cylinder in marine turbines varies between 150 pounds and 200 pounds per square inch, and to go much beyond this, it seems, is of no special advantage. (Note that Parsons turbine is dealt with in this article.) The reason will be found due to increased dimensions and additional weight and space without a correspondingly large actual increase in economy, although the theoretical gain is quite apparent. The probability of dangerous distortions in the turbine cylinders, unless made excessively heavy, caused by high temperature differences, when exsisting within the same casing, is reason enough to exclude higher pressures. Vacuum is carried usually between $27\frac{1}{2}$ inches and $28\frac{1}{2}$ inches, but rarely below or above, because of the loss thermodynamically on the one hand and the vary rapid increase of steam volume on the other, and therefore the necessity for larger blade passages. The length and angle of the blade in the exhaust end of the turbine, together with other details, act to some extent as limiting factors with respect to vacuum.

Preliminary Design.—An approximate outlay must be made first with the turbines running at such minimum speed as will permit their construction on the allowance of weight and space allotted to this part of the machinery installation. If weight and space are only of secondary importance, then the turbine speed may be developed entirely from dimensions and speed most suitable to the propeller. Considerations of subdivisions within the ship, facilities for locating line shafting, handling parts during repairs and for operation while running, to some extent determine arrangement and number of turbines. This

latter item, however, is largely a result of propeller, or its capacity to absorb the power transmitted to it, on a diameter suitable to its speed. The chance is very slight, indeed, of procuring a successful marine turbine outfit without at the same time, in detail, considering the propeller. On this account a method of propeller design is taken up in connection with the general proposition of marine turbine design.

THE PROPELLER.

The theory and design of propellers will be found comprehensively discussed in the following works: "Resistance of Ships and Screw Propellers," by D. W. Taylor, "Naval Architecture," by Cecil H. Peabody, "Marine Propellers," by S. W. Barnaby, and in various papers presented by Prof. W. F. Durand, Naval Constructor D. W. Taylor and others.

While in the matter appended below nothing new is introduced so far as the general subject goes, the application of the methods in combination with data relating to turbine-driven propellers becomes interesting, and consists in establishing preliminary dimensions of the propeller by limitation in tip speed and pressure on the blade surface, using the dimensions so obtained to determine the thrust for given power conditions and speeds of advance.

Dimensions of propellers are influenced directly by—

1. Speed (K) of ship, in knots per hour.
2. The effective horsepower ($E.H.P.$) needed for propulsion.
3. Real slip (S).
4. Diameter ratio $\left(\frac{D}{P}\right)$ which is the relation of diameter and pitch.
5. Number (N) of blades.
6. Pressure Ps per unit of projected blade surface.

Real slip of propeller $S = S_1 (1 - w) + w$,

in which

S_1 = apparent slip.

w = wake.

The effective horsepower delivered by a propeller per second is obtained from

$$E.H.P. = \frac{1}{550 \times 60^3} P^3 \times R^3 \times D^2 \times N(1-S)(aSA - fB) \quad . \quad (1)$$

The total horsepower absorbed by a propeller per second from

$$H.P. = \frac{1}{550 \times 60^3} P^3 \times R^3 \times D^2 \times N(aSA + fC), \quad . \quad (2)$$

Dividing (1) by (2) we get the propeller efficiency (e)

$$e = (1 - S)\frac{aSA - fB}{aSA + fC}$$

The blade characteristics A, B and C vary with contour of blade and diameter ratio $\frac{D}{P}$, and their value, for unit width ratio, will be found tabulated in full in the works by Taylor and Peabody previously referred to, and below for the case in hand.

For

$\frac{D}{P} = 1.00$ in a Taylor blade.

$A = 0.565$,

$B = 0.759$,

$C = 3.580$,

a, known as the thrust factor is for three-bladed screws $= 3.3 - .4\frac{D}{P}$,

f, co-efficient of friction, $= .016$.

The values of A, B and C given, having been determined for a maximum width ratio of unity, must, before inserted in the formulæ, be multiplied by the width ratio $\frac{L}{D}$ found to exist in the propeller to be used. Different values of diameter ratio and slip should be tried until a diameter is obtained suited for the revolutions, bearing in mind that a small diameter ratio gives a large diameter and slow revolutions, while a large diameter ratio gives the reverse.

The diameter ratio in screw propellers operated by Parsons turbines, as now made, are confined within narrow limits, usually from .9 and 1.1, with an average mean of 1.0.

Experiments by Thornycroft and Barnaby, as well as late trials, have shown that 11½ pounds pressure per square inch of projected blade surface, with a tip submersion of 12 inches, was about as much as could be sustained before formation of cavities occurred. A slight increase of pressure for additional tip submersion may be allowed. Judgment in the choice of blade section must also be exercised, as with certain shapes the permissible pressure becomes considerably less than 11 to 12 pounds.

This limit having to be observed, it becomes necessary to use an extraordinary wide blade, the diameter being limited by other considerations. However, even the width of the blade has a limit beyond which no additional thrust is transmitted, regardless of area ratio, and was found, in experiments performed by Prof. Durand with four-bladed screws, to occur when the helicoidal area reached 63 per cent. of the disc. Actual cases exist, however, with three-bladed turbine-driven screw propellers where this ratio is as high as 68 per cent., and in most instances exceeds 63 per cent., while with engine-driven screws more than 45 per cent. is rarely ever used. It is also a well-known fact that the frictional resistance of a body driven through the water is enhanced by an increasing power of the speed, and the power absorbed in accordance. The tip speed of the propeller must therefore be kept within defined limits, which, in turbine-driven screws, usually are within 10,000 feet per minute. In engine-driven screws the rotational speed is considerably less, and, in consequence thereof, the losses diminished and the efficiency increased.

The proposed method resolves itself thus in determining first the diameter from tip speed and pressure per unit of blade surface, bearing in mind that the blade area should not exceed 66 to 68 per cent. of the disc and that the slip ratio should be low. And then, having approximately defined diameter, revolutions, slip and blade area, by the use of formulæ (1), to see

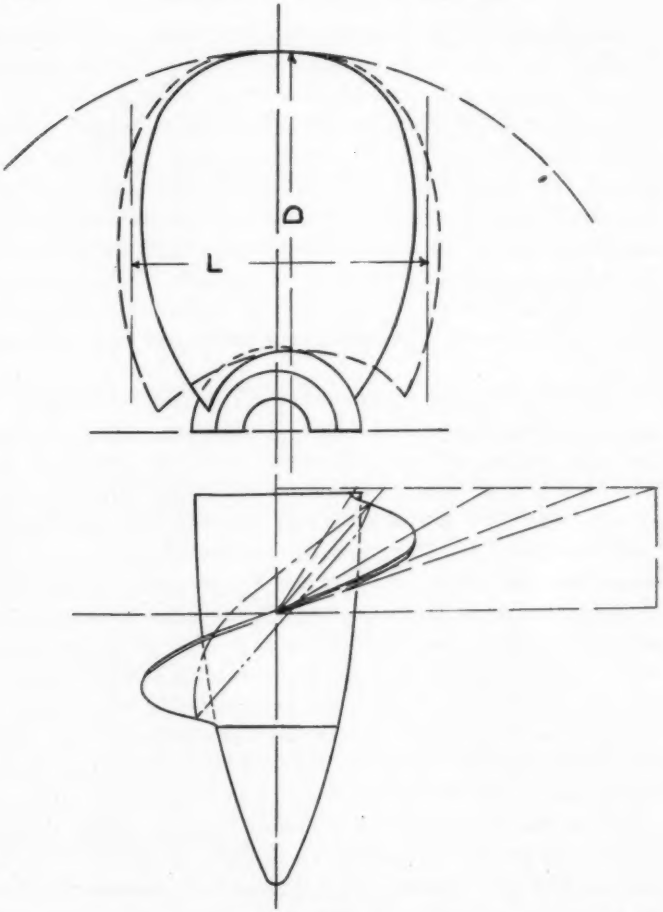


Figure 1.

Diameter = 4 feet 9 inches.
Pitch = 4 feet 9 inches.
R.P.M. = 650.
Disc area ratio = .63.
Projected area = 9.2 feet.
Mean width ratio = $\frac{L}{D} = \frac{1.9}{4.75} = .4$.
Three blades.

if these quantities when inserted in said formulæ, will deliver the horsepower. To illustrate the foregoing we will assume:

$E.H.P.$ = 3,900, to be equally divided among three shafts;

R = 650, determined by minimum speed of revolution per minute of turbine rotor from approximate outlay and having reference to weight and space and assumed steam speeds;

Speed of ship = $27\frac{1}{2}$ knots per hour;

Real slip = 15.4 per cent. from an assumed apparent slip = 10 per cent. and wake = 6 per cent;

$\frac{L}{D}$ = .4 from drawing made of the preliminary propeller, see Figure 1, the dimensions of which have been approximated from maximum tip speed and pressure per unit of surface;

$\frac{D}{P}$ = 1, for a trial;

N = 3, by choosing a three-bladed screw.

Effective thrust, per screw,

$$T = \frac{E.H.P.}{K} \times 325.8 = \frac{\frac{1}{3} \times 3{,}900 \times 325.8}{27.5} = 15{,}400 \text{ pounds.}$$

Helicoidal area = (Disc — Hub) × disc area ratio.

The hub is approximated from shaft, easily obtained when power and revolutions are known.

Assume disc area ratio = .63 = M;

tip speed = 9,800 feet per minute = T_S;

revolutions per minute = 650 = R;

area of hub = 1 square foot = H in square feet.

Helicoidal area either by trial or some such simple formulæ as this: $\left\{ \frac{1}{12.5}\left(\frac{T_S}{R}\right)^2 - H \right\} \times M = \left(\frac{9{,}800^2}{12.5 \times 650^2} - 1\right) \times .63 =$ $(18.2 - 1) \times .63 = 10.8$ square feet.

Projected area, where $\frac{D}{P} = 1$, $= 9.2$ square feet, either from drawing or well-known formulæ.

Pressure per square inch of projected surface $Ps = \frac{15{,}400}{9.2 \times 144}$ $= 11.6$ pounds.

The disc being 18.2 square feet gives a diameter of 4 feet 9 inches.

Getting out a propeller substantially as found by the foregoing, and using the dimensions in connection with formulæ (1), we are now able to determine the effective horsepower which it will deliver, or if the power is known, we may find the diameter for slip ratio and diameter ratio assumed, using the E.H.P. already stipulated.

Applied to our example we get:

$$E.H.P. = \frac{3{,}900}{3} = 1{,}300 \text{ on each shaft;}$$

S (real slip) $= 15.4$ per cent.;

S_1 (apparent slip) $= 10$ per cent.;

$$P \times R = 3{,}095 \text{ from } \frac{101.3\ K}{1 - S_1} = P \times R;$$

$$a = 3.3 - .4 \times 1 = 2.9.$$

Blade characteristics from table

$$A = .565 \times .4 = .226,$$
$$B = .759 \times .4 = .3036,$$
$$C = 3.58 \times .4 = 1.432,$$

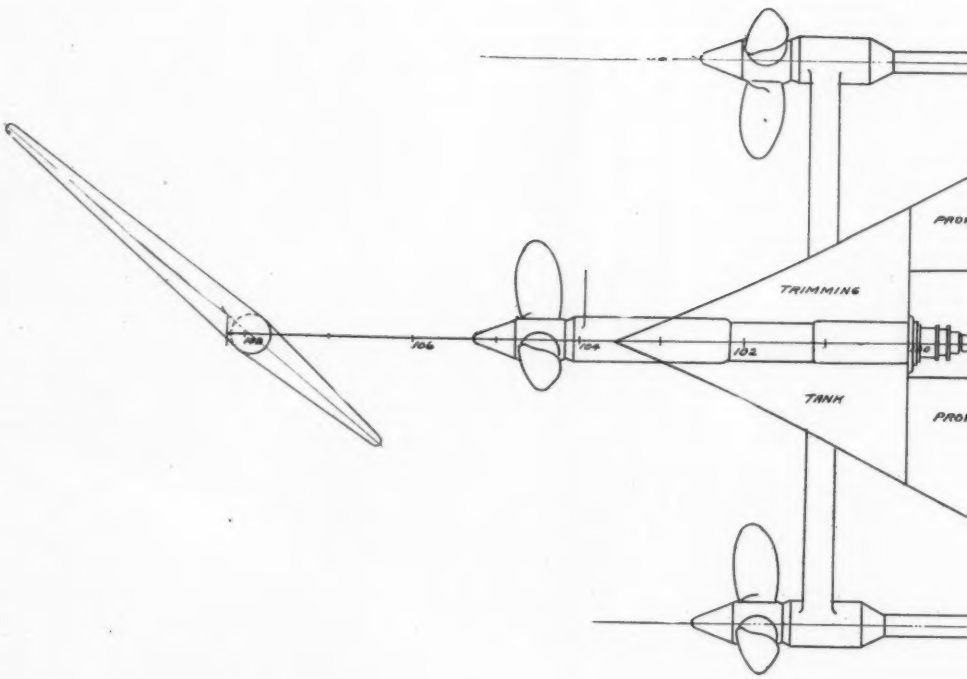
$$1{,}300 = \frac{3{,}095^3}{550 \times 60^3} \times D^2 \times 3\,(1 - .154)$$
$$(2.9 \times .154 \times .226 - .016 \times .3036),$$

$$D = \sqrt{\frac{1{,}300 \times 550 \times 60^3}{3{,}095^3 \times .2436}} = \sqrt{22.5} = 4 \text{ feet } 9 \text{ inches.}$$

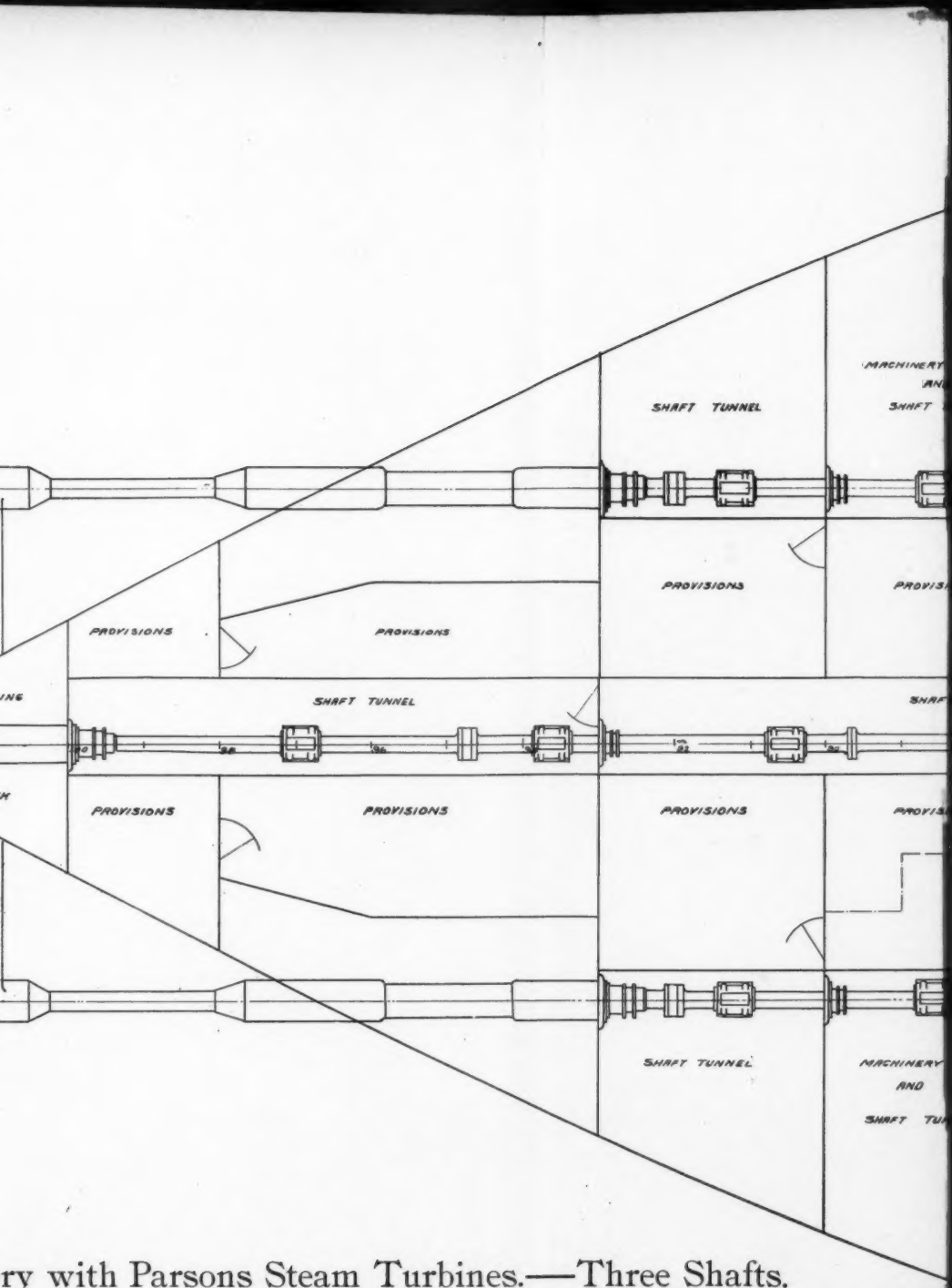
The diameter ratio is assumed equal 1; therefore, $\frac{4.75}{P} = 1$, and $P = 4.75$ feet.

$$R \text{ (as before)} = \frac{3{,}095}{4.75} = 650.$$

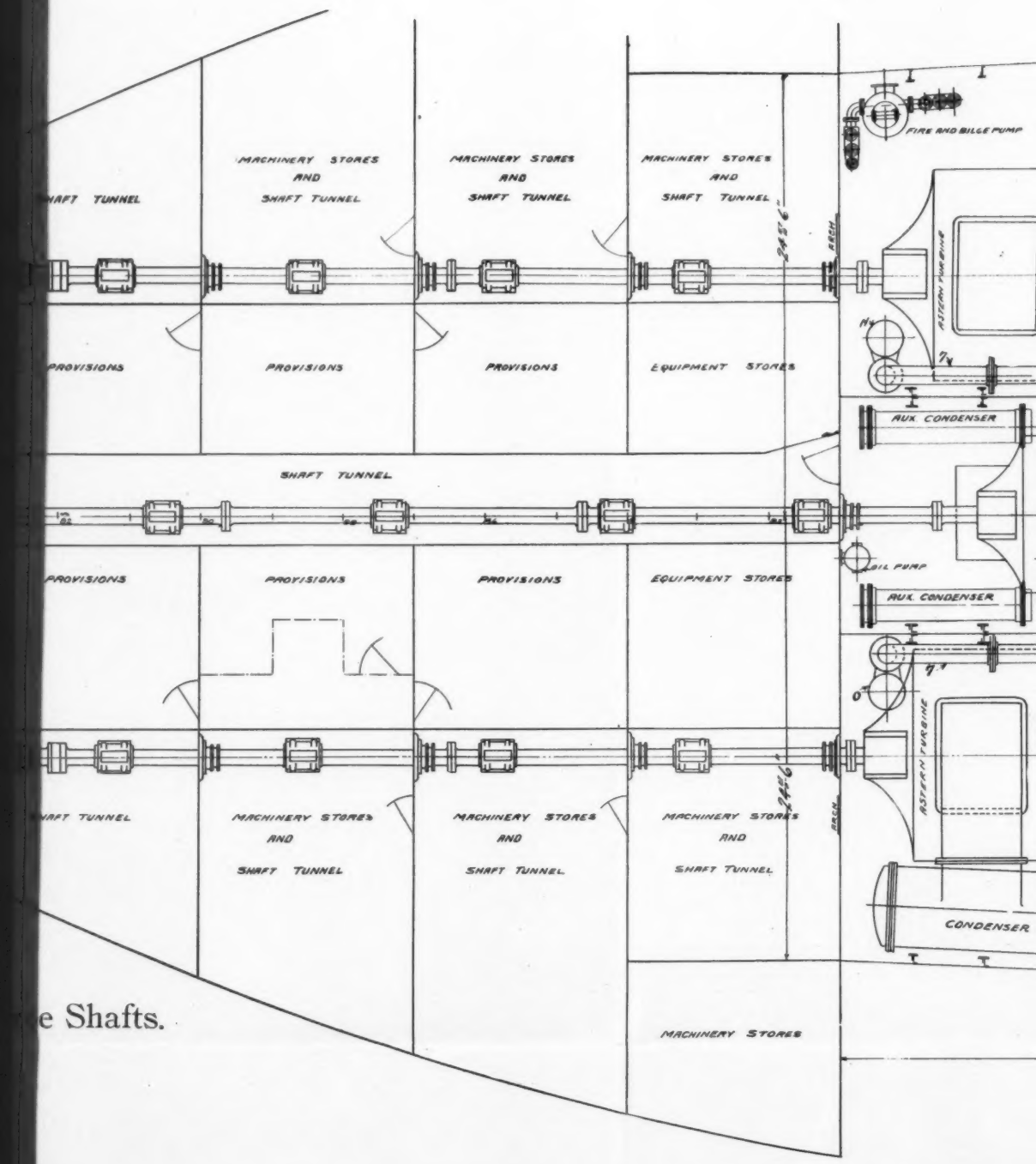
$$\text{Propeller efficiency} = \frac{.846 \times .095}{0.1239} = 64.8 \text{ per cent.}$$



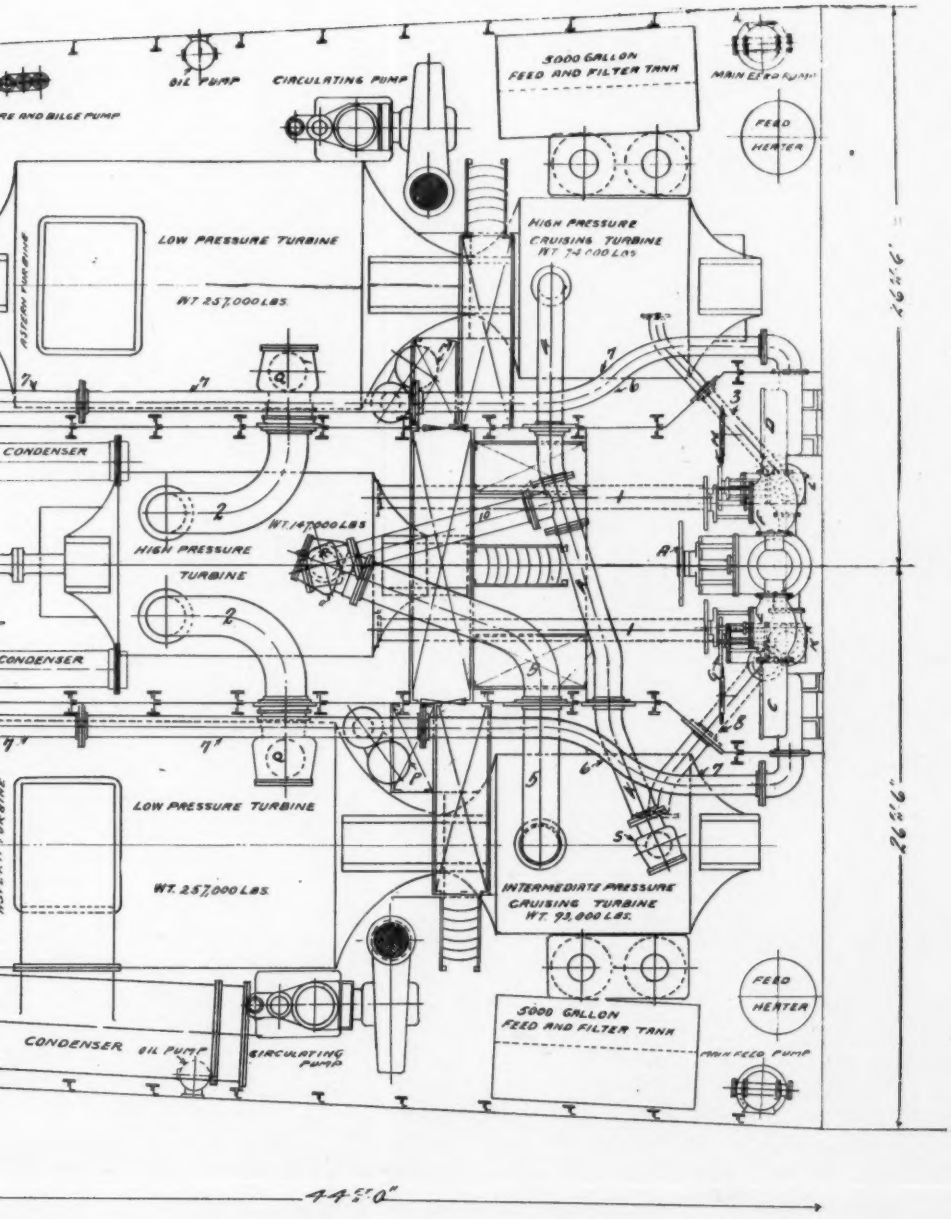
Arrangement of Machinery with



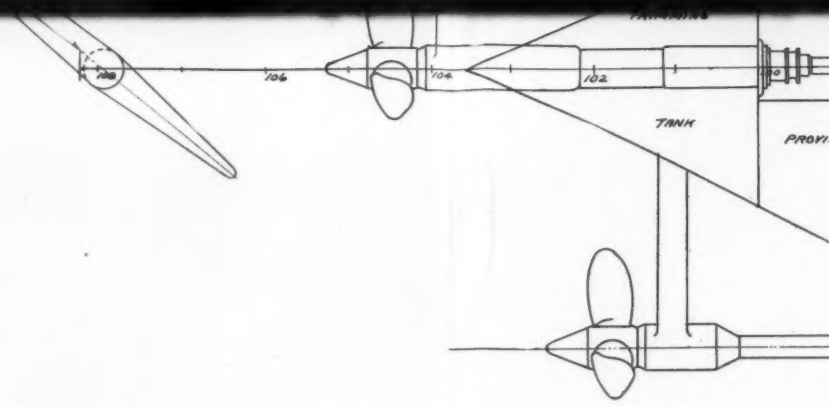
ry with Parsons Steam Turbines.—Three Shafts.


SHAFT TUNNEL
MACHINERY STORES AND SHAFT TUNNEL
MACHINERY STORES AND SHAFT TUNNEL
MACHINERY STORES AND SHAFT TUNNEL
24'5" 6"
ARCH
FIRE AND BILGE PUMP
ASTERN TURBINE
PROVISIONS
PROVISIONS
PROVISIONS
EQUIPMENT STORES
AUX. CONDENSER
SHAFT TUNNEL
OIL PUMP
PROVISIONS
PROVISIONS
PROVISIONS
EQUIPMENT STORES
AUX. CONDENSER
ASTERN TURBINE
SHAFT TUNNEL
MACHINERY STORES AND SHAFT TUNNEL
MACHINERY STORES AND SHAFT TUNNEL
MACHINERY STORES AND SHAFT TUNNEL
24'6"
ARCH
CONDENSER
MACHINERY STORES

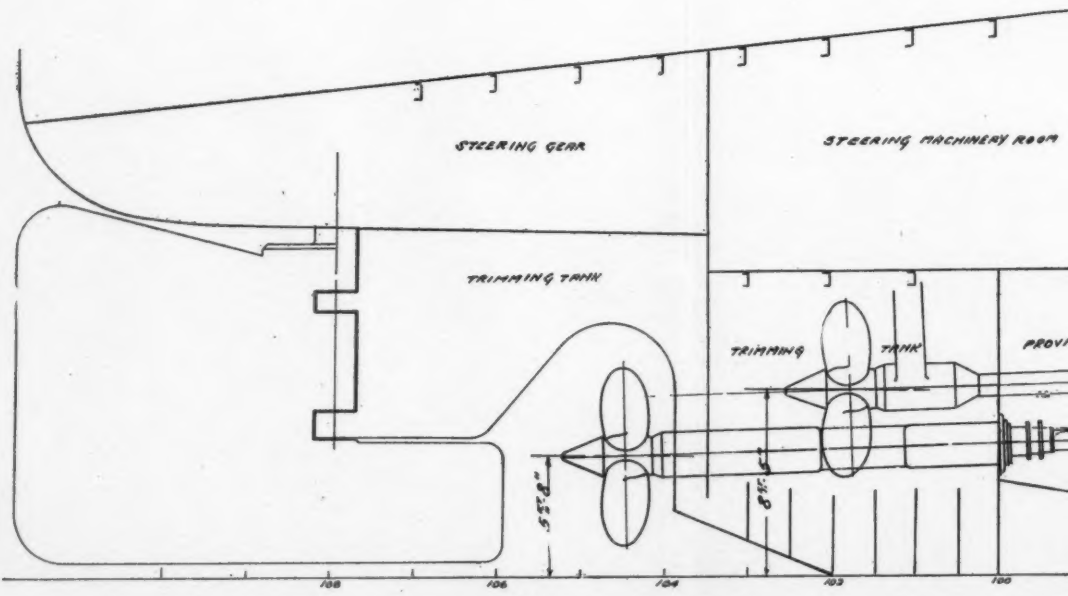
e Shafts.

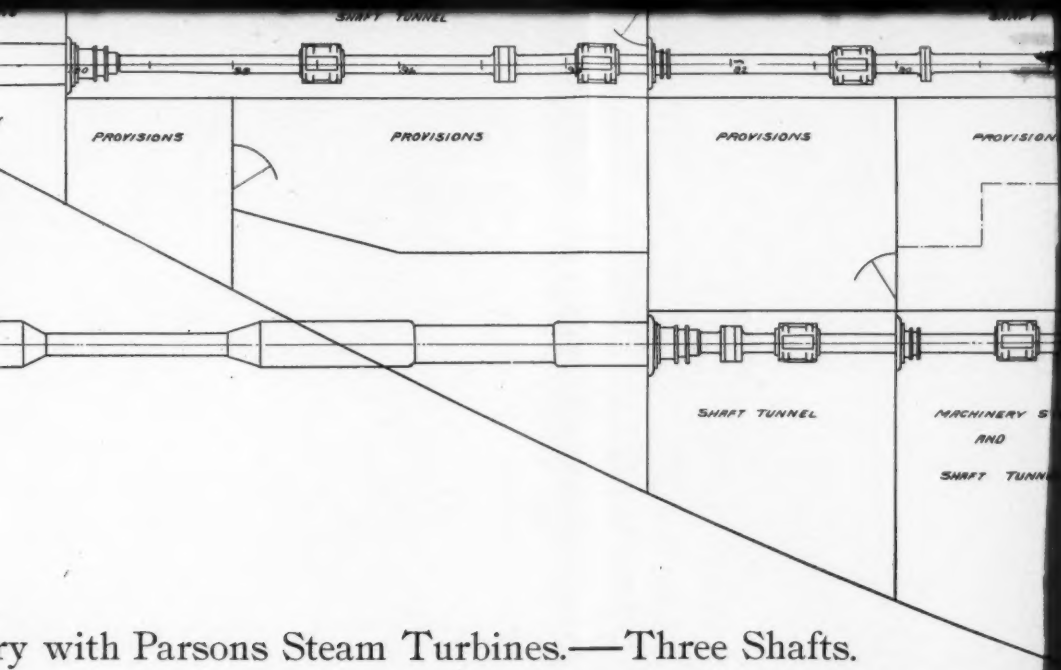


AIR SHAFT
FOR CENTRE MACHINERY SPACE



Arrangement of Machinery with





ry with Parsons Steam Turbines.—Three Shafts.

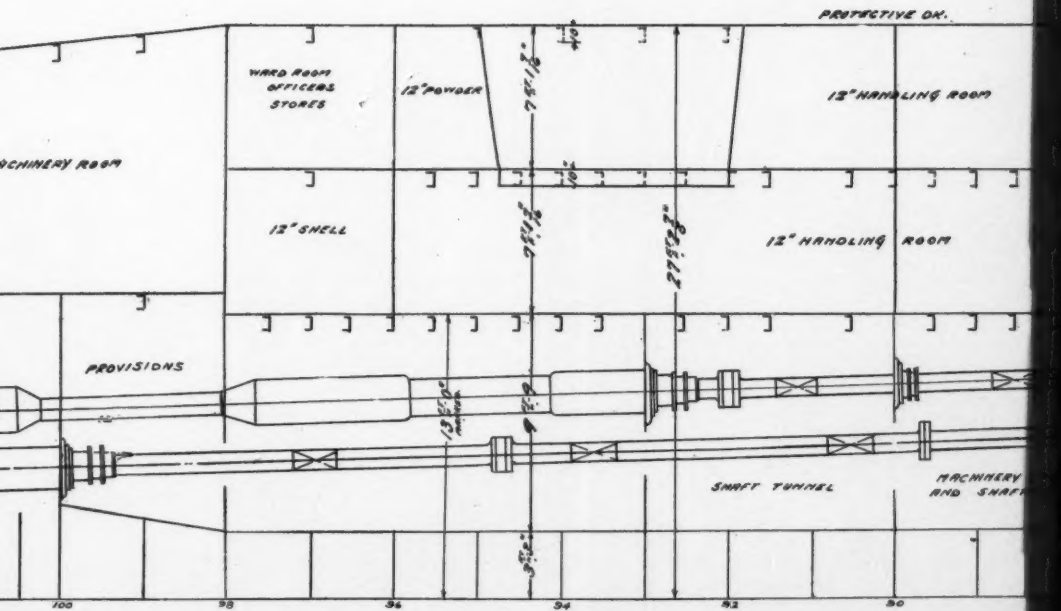
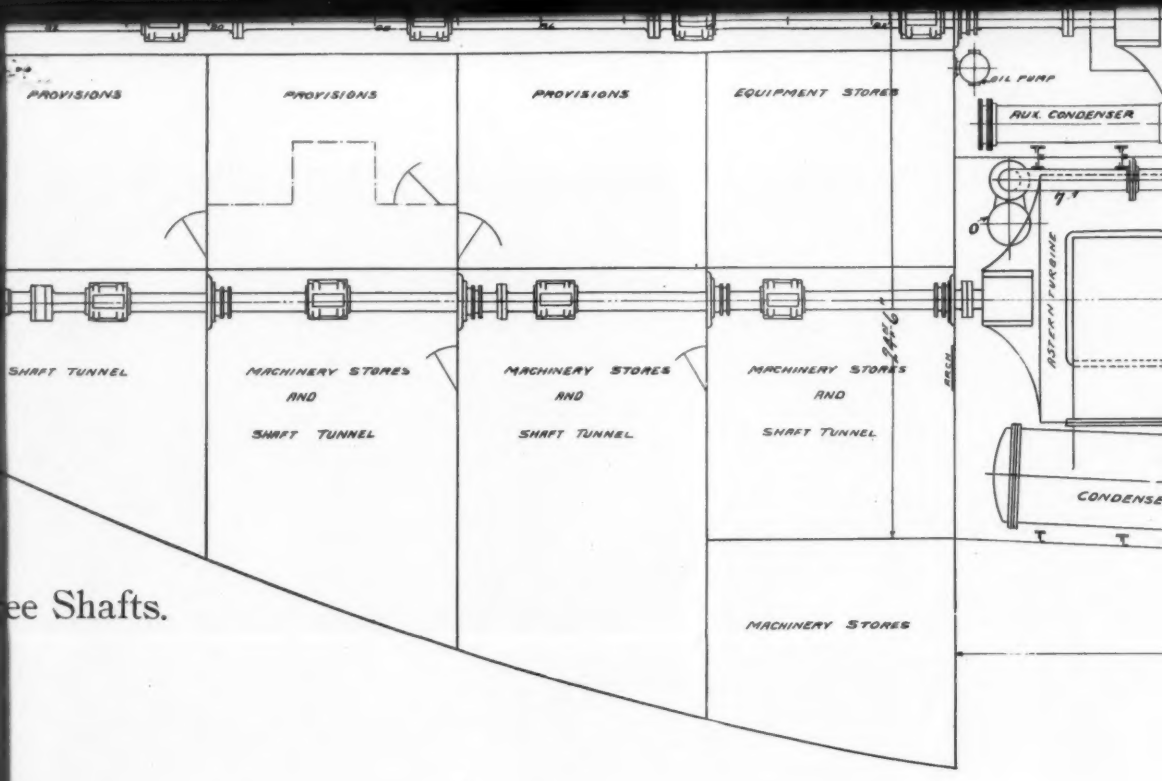


Plate I.



ee Shafts.

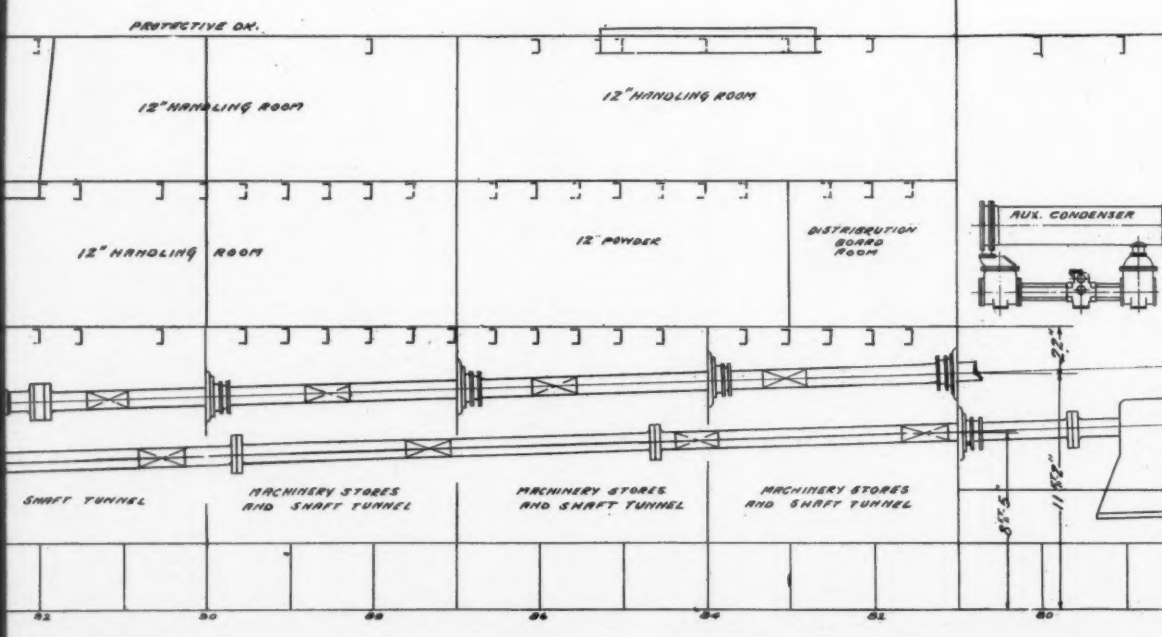
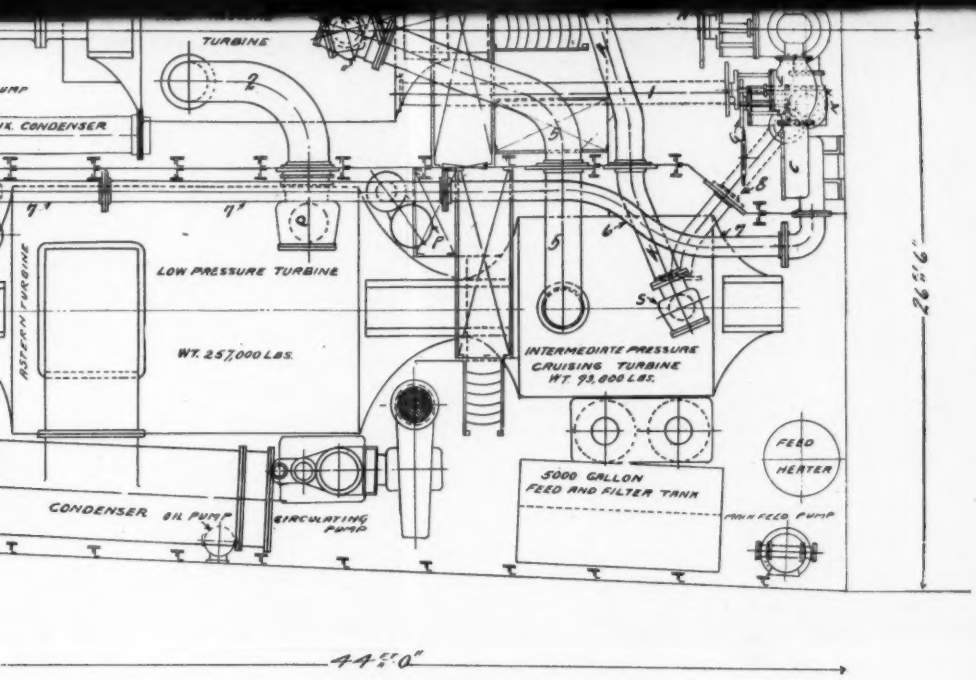
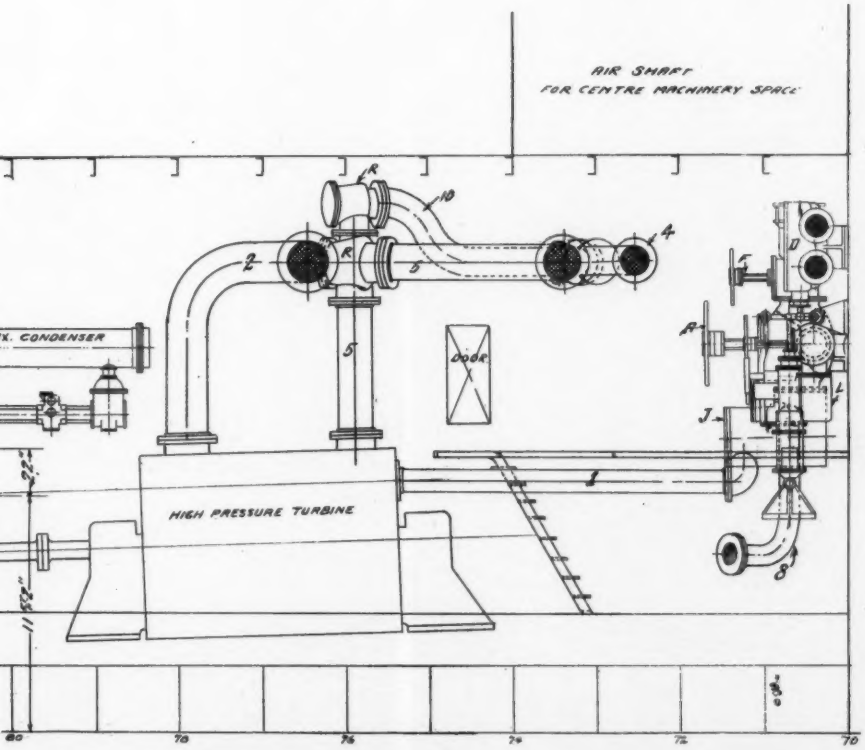
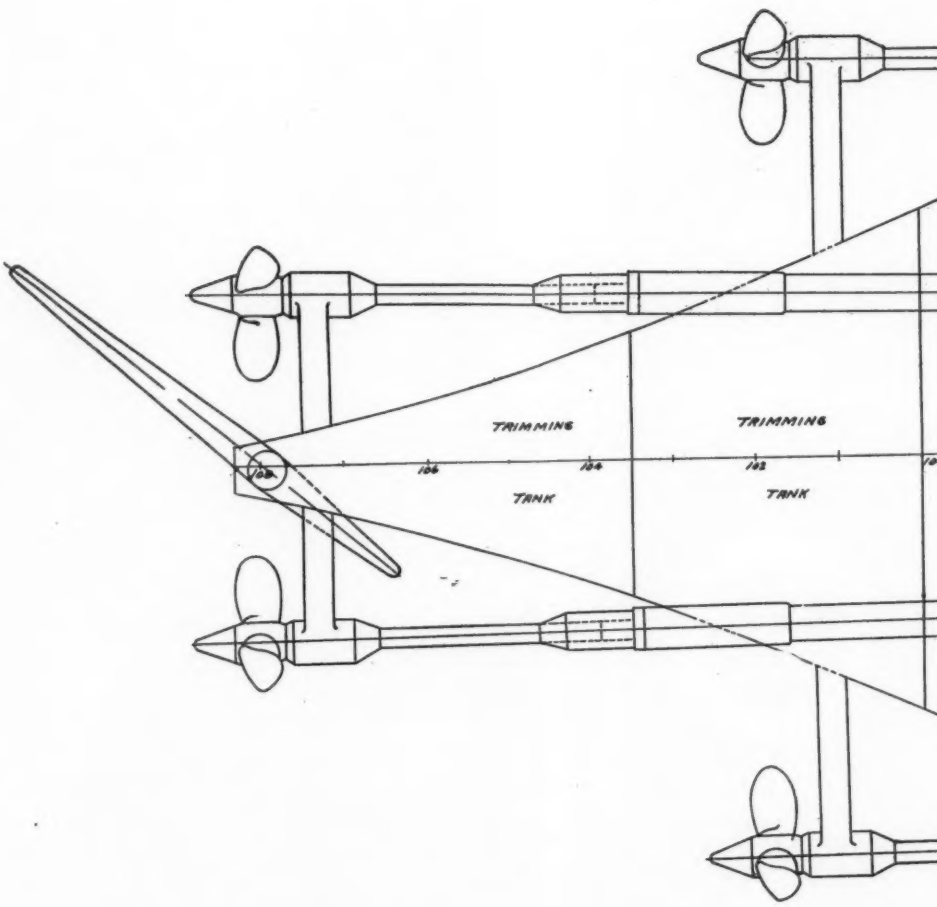


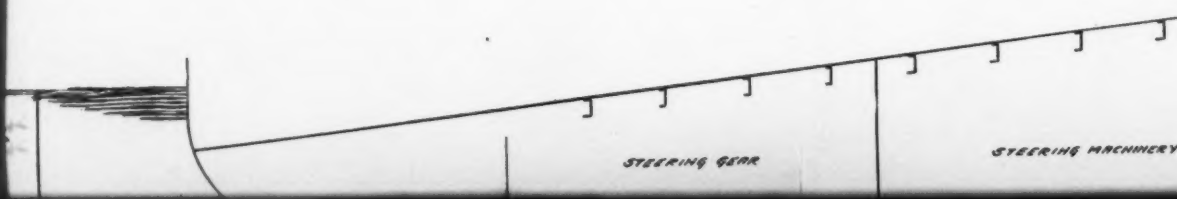
Plate I.

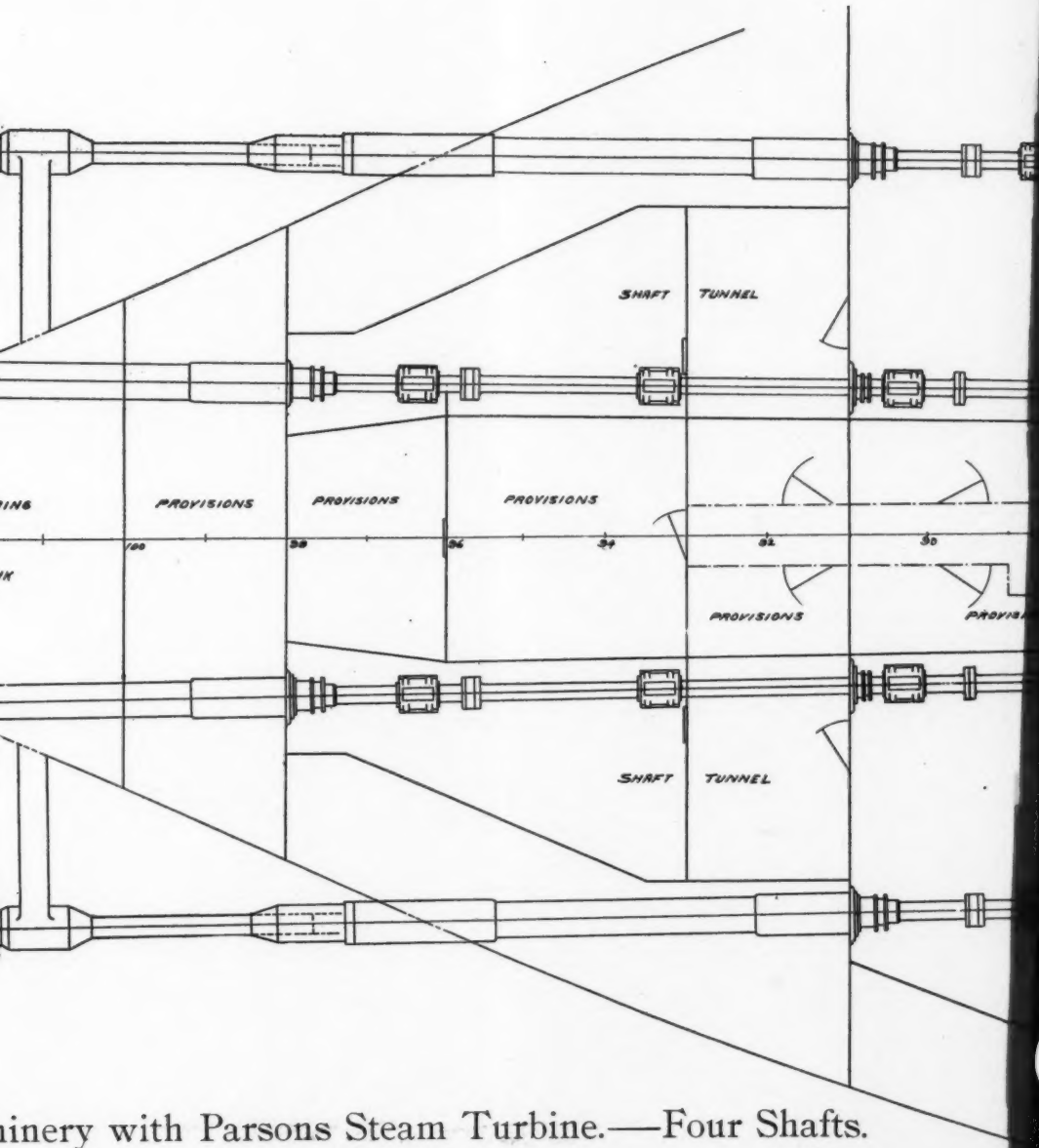

TURBINE
LOW PRESSURE TURBINE
WT. 257,000 LBS.
INTERMEDIATE PRESSURE
CRUISING TURBINE
WT. 93,000 LBS.
5000 GALLON
FEED AND FILTER TANK
FEED
HEATER
CONDENSER
OIL PUMP
CIRCULATING
PUMP
MAIN FEED PUMP
44 FT. 0"


AIR SHAFT
FOR CENTRE MACHINERY SPACE
DOOR
HIGH PRESSURE TURBINE

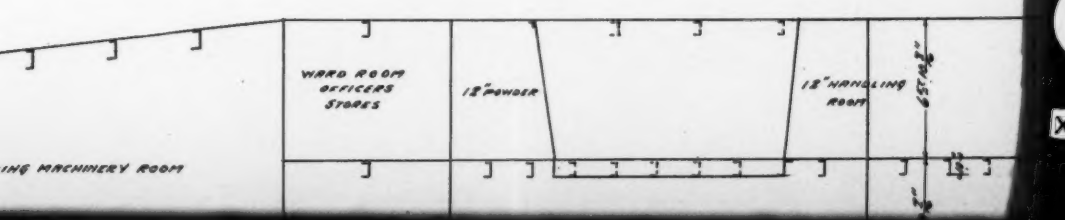


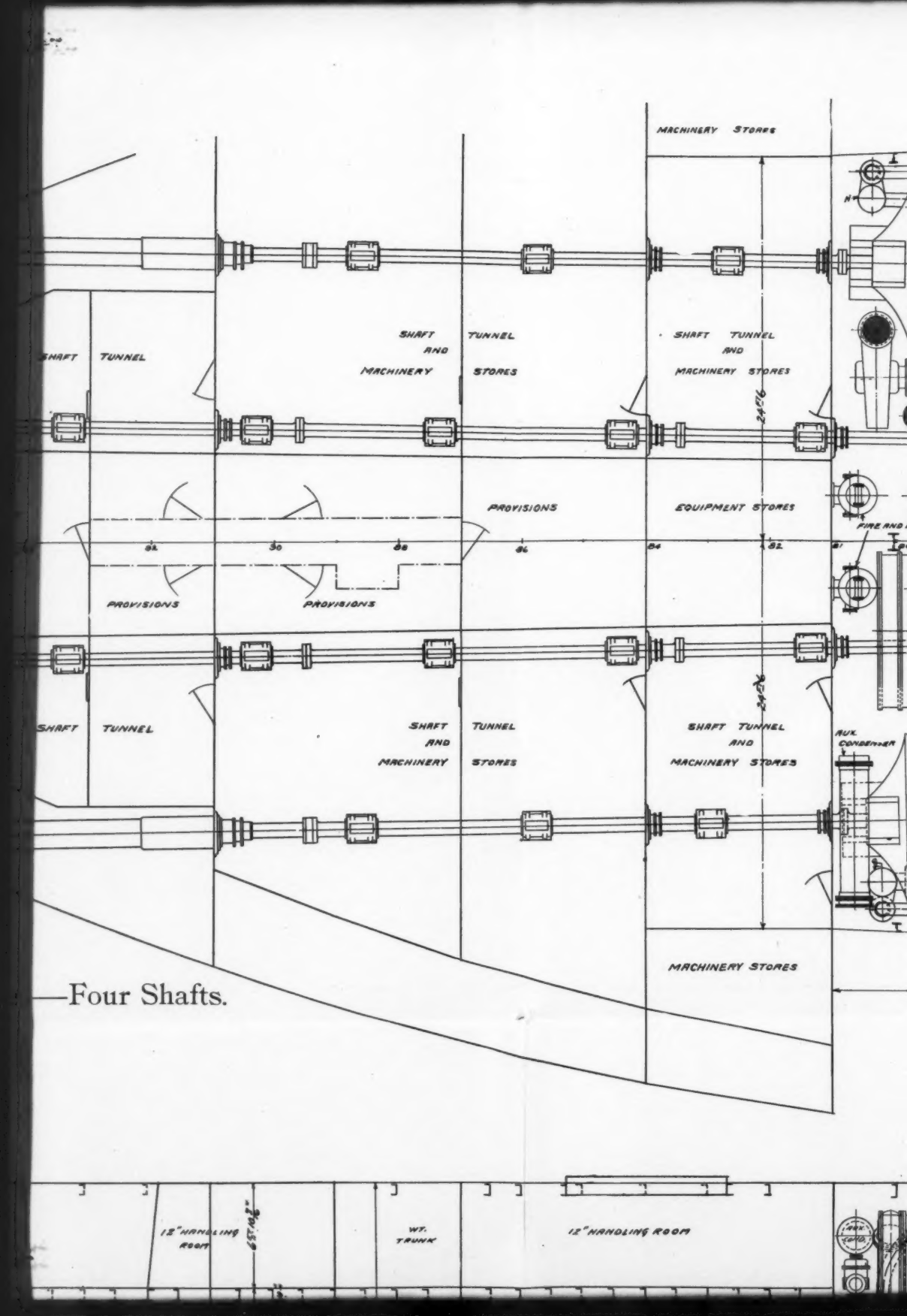
Arrangement of Machinery w



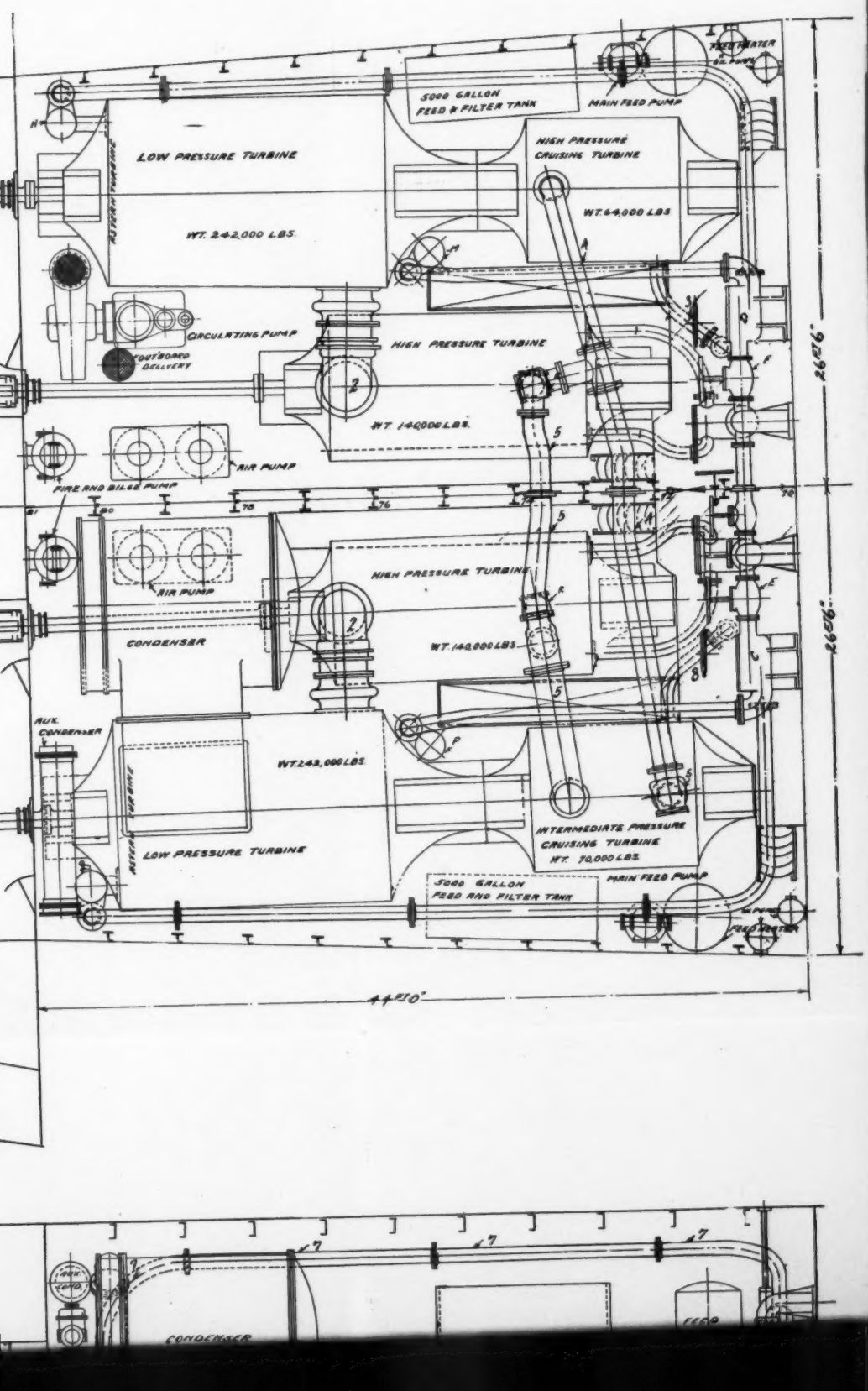


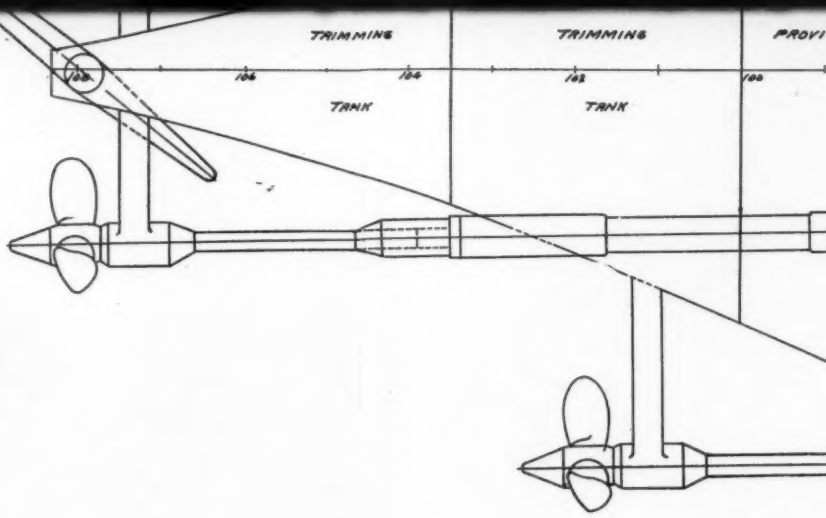
hinery with Parsons Steam Turbine.—Four Shafts.



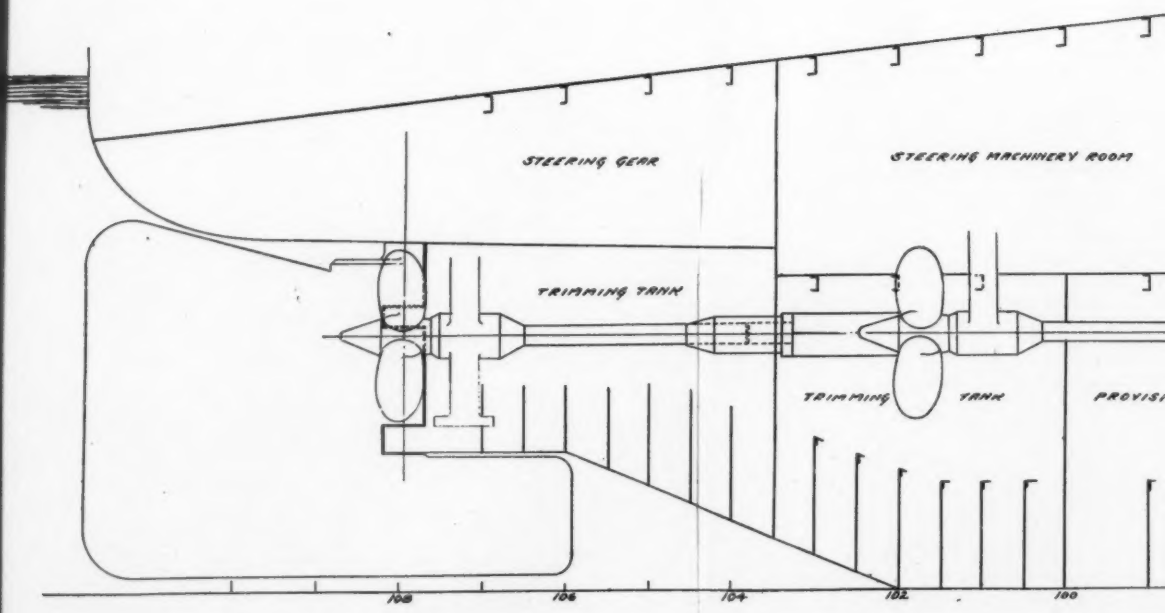

MACHINERY STORES
SHAFT TUNNEL
SHAFT TUNNEL AND MACHINERY STORES
SHAFT TUNNEL AND MACHINERY STORES
PROVISIONS
EQUIPMENT STORES
PROVISIONS
PROVISIONS
SHAFT TUNNEL
SHAFT TUNNEL AND MACHINERY STORES
SHAFT TUNNEL AND MACHINERY STORES
AUX. CONDENSER
MACHINERY STORES
12" HANDLING ROOM
WT. TRUNK
12" HANDLING ROOM

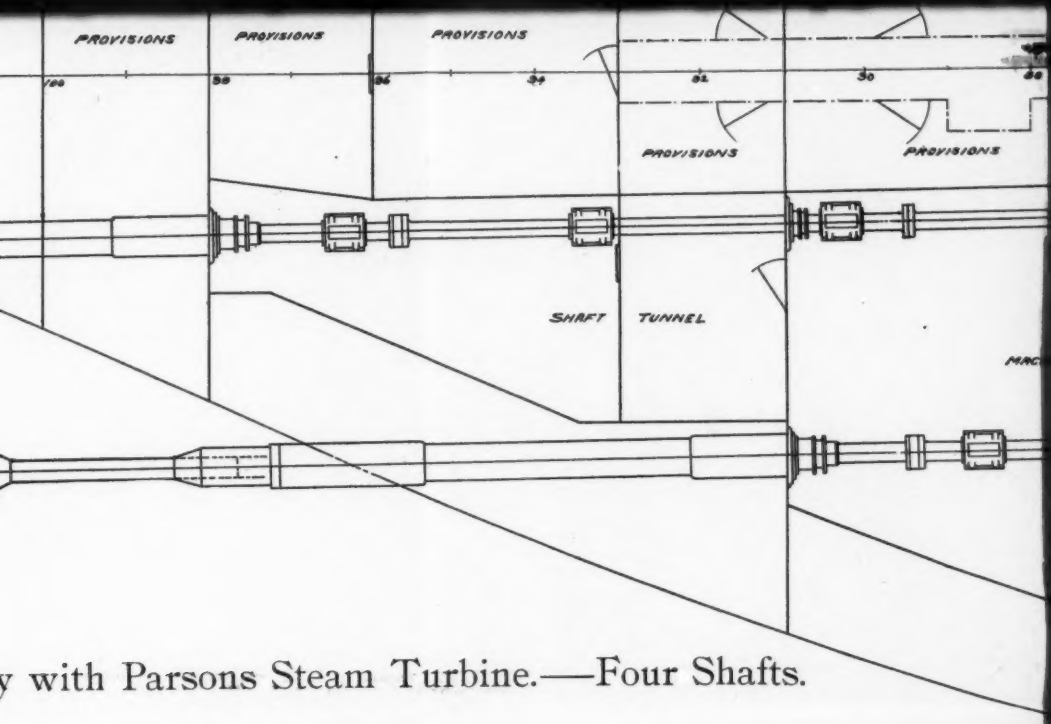
—Four Shafts.


LOW PRESSURE TURBINE
WT. 242,000 LBS.
ASTERN TURBINE
5000 GALLON FEED & FILTER TANK
MAIN FEED PUMP
FEED WATER H.P. PUMP
HIGH PRESSURE CRUISING TURBINE
WT. 64,000 LBS
CIRCULATING PUMP
OVERBOARD DELIVERY
HIGH PRESSURE TURBINE
WT. 140,000 LBS.
AIR PUMP
FIRE AND BILGE PUMP
AIR PUMP
CONDENSER
HIGH PRESSURE TURBINE
WT. 140,000 LBS.
AUX. CONDENSER
WT. 243,000 LBS.
ASTERN TURBINE
LOW PRESSURE TURBINE
INTERMEDIATE PRESSURE CRUISING TURBINE
WT. 70,000 LBS.
5000 GALLON FEED AND FILTER TANK
MAIN FEED PUMP
FEED WATER H.P. PUMP
26'6"
26'6"
44'10"
CONDENSER



Arrangement of Machinery with I





y with Parsons Steam Turbine.—Four Shafts.

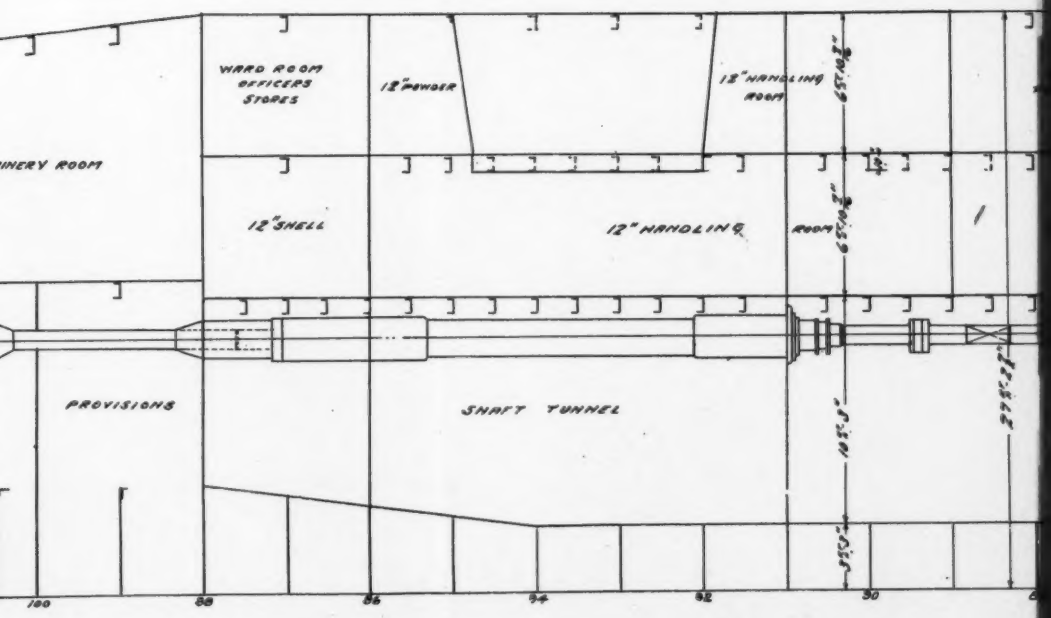
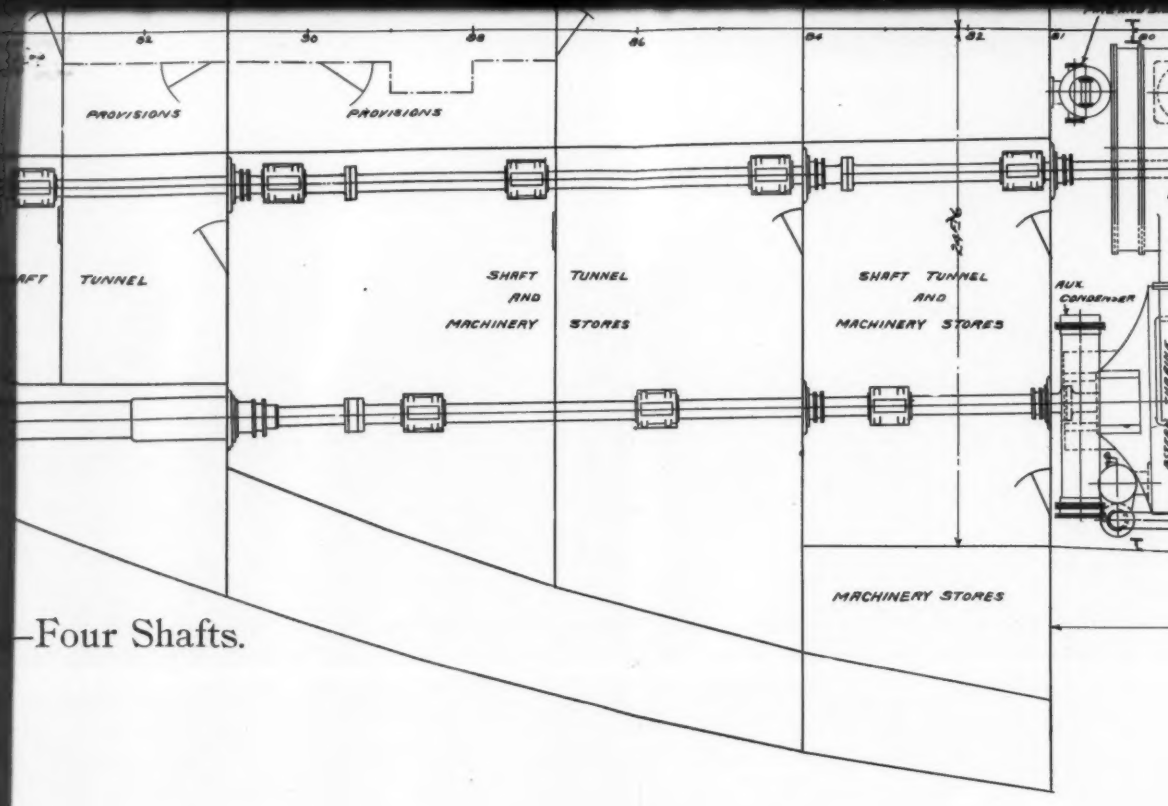


Plate II.



—Four Shafts.

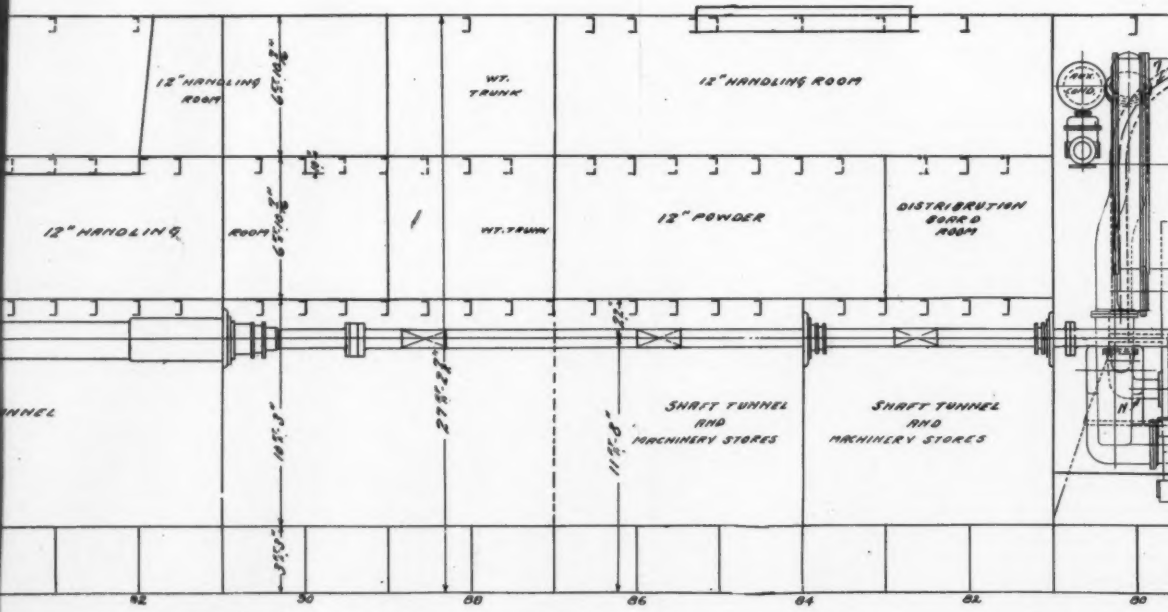
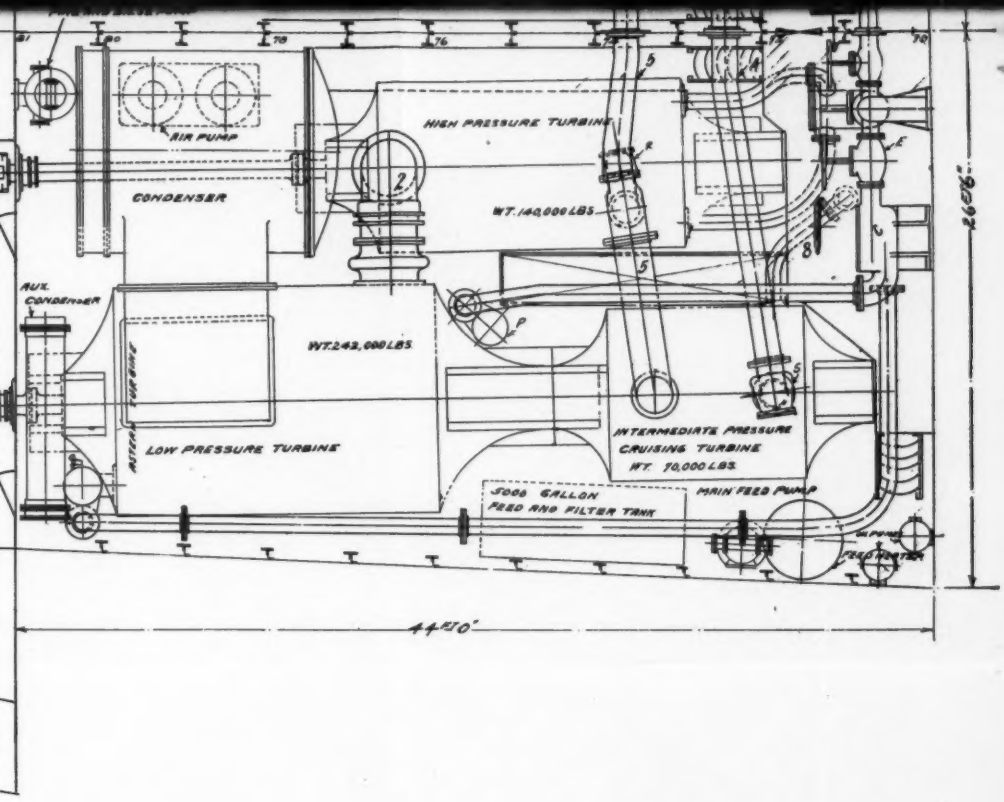
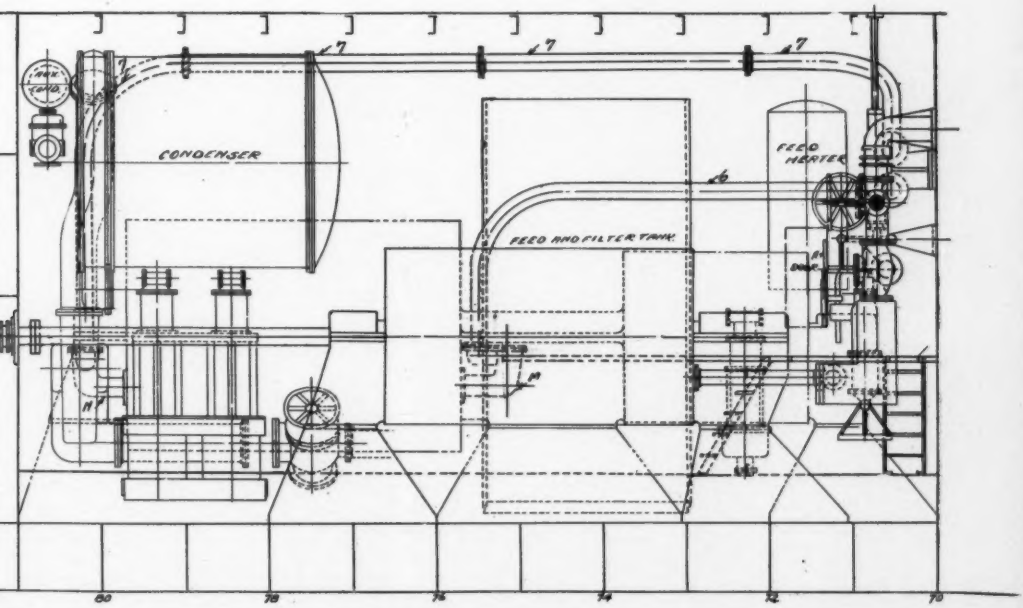


Plate II.


AIR PUMP
CONDENSER
HIGH PRESSURE TURBINE
WT. 140,000 LBS.
AUX. CONDENSER
WT. 243,000 LBS.
LOW PRESSURE TURBINE
INTERMEDIATE PRESSURE CRUISING TURBINE
WT. 70,000 LBS.
5000 GALLON FEED AND FILTER TANK
MAIN FEED PUMP
26'6"
44'10"


CONDENSER
FEED HEATER
FEED AND FILTER TANK
80
78
76
74
72
70

Total brake horsepower at turbine end of shafting = $\frac{1.05 \times 1,300}{.648 \times .97} = 2,170$ based on 64.8 per cent. propeller efficiency and 97 per cent. line-shaft efficiency and 5 per cent. increase due to the influence of thrust deduction and wake.

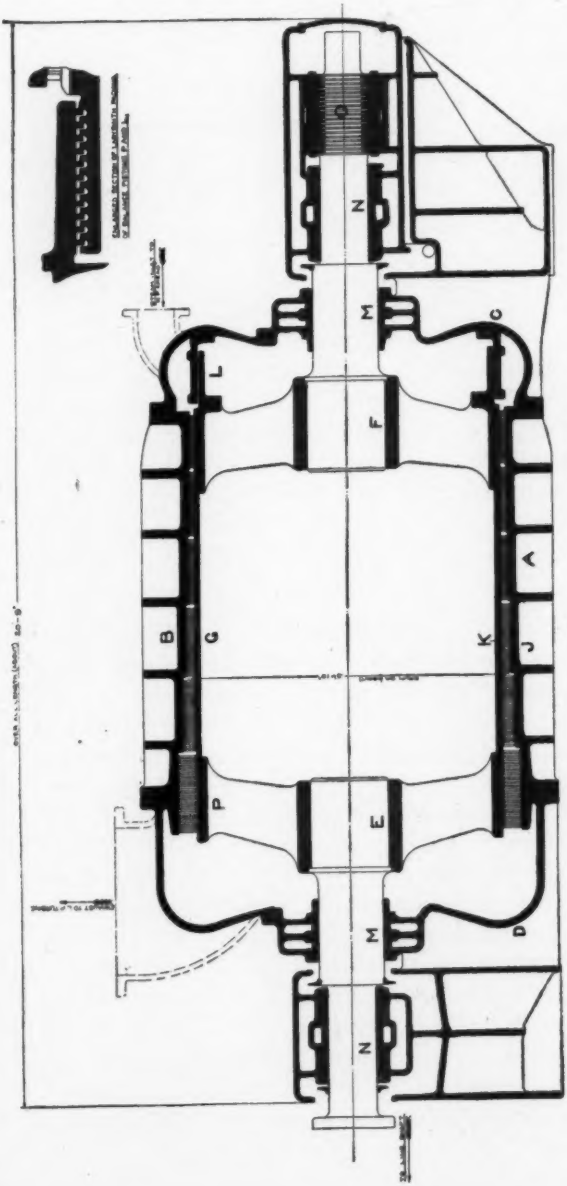
The indicated horsepower of a reciprocating engine to deliver foregoing effective horsepower would be $\frac{2,170}{.9} = 2,411$, based on 90 per cent. mechanical efficiency. Should the results obtained by the two methods not agree, it becomes necessary to give different values for the quantities entering, principally depending upon diameter ratio and slip, until a close agreement is obtained. This is readily ascertained by trial of different diameter and slip ratios for the proposed screw.

The procedure indicated may appear circumstantial and tedious, but if executed carefully, will give practicable results, and, moreover, avoids the uncertainty that various other methods involve.

THE TURBINE.

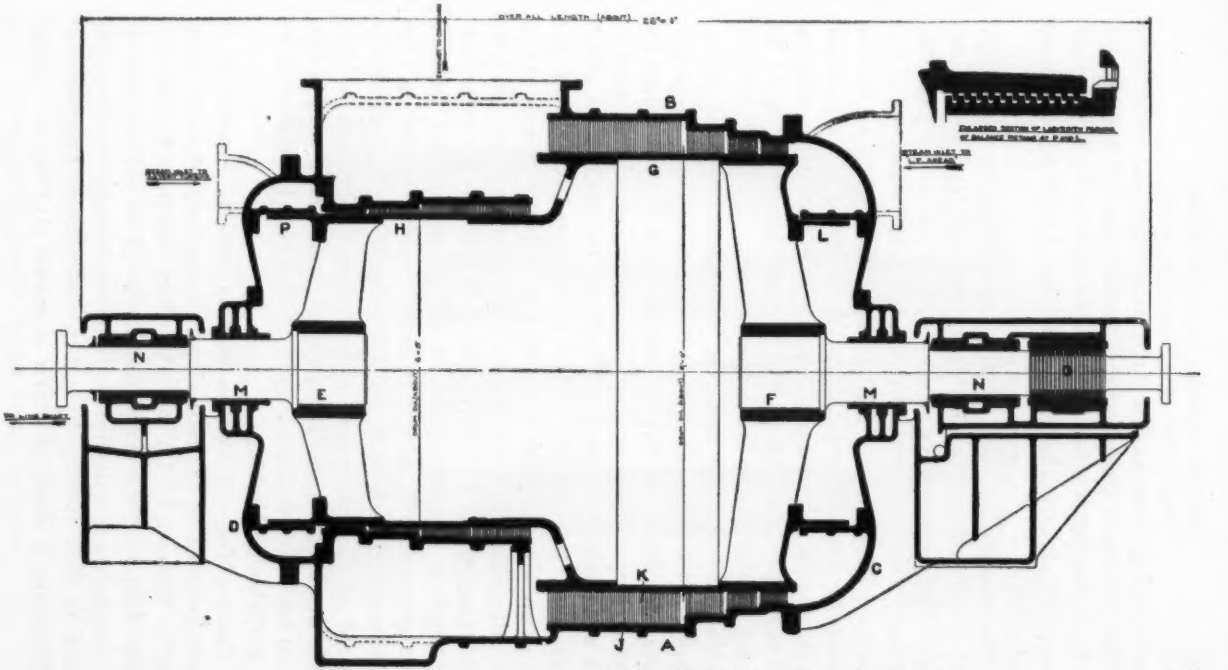
Arrangement.—Installations with Parsons turbine when used for marine purposes comprise arrangements with either three or four shafts, each usually having one propeller. The choice is more due to conditions within the ship, such as subdivisions of compartments, than to mechanical necessities in the turbine. In very large installations, however, where the power is extraordinary, four shafts meet the functions of both screw and turbine to better advantage. With three shafts, Plate I, the high-pressure turbine is always placed with its shaft in the center, while each of the low-pressure turbines drives an outboard shaft.

With four shafts, Plate II, either both high-pressure turbines drive the inboard shafts and the low-pressure turbines the outboard, or this is reversed. The backing turbines are always placed in the low-pressure end of the low-pressure turbines. Separate backing turbines could also be attached to the high-pressure shafts should they be deemed essential for



ALLIS-CHALMERS-PARSONS HIGH PRESSURE TURBINE.

Plate III.



ALLIS-CHALMERS-PARSONS LOW PRESSURE AND ASTERN TURBINE

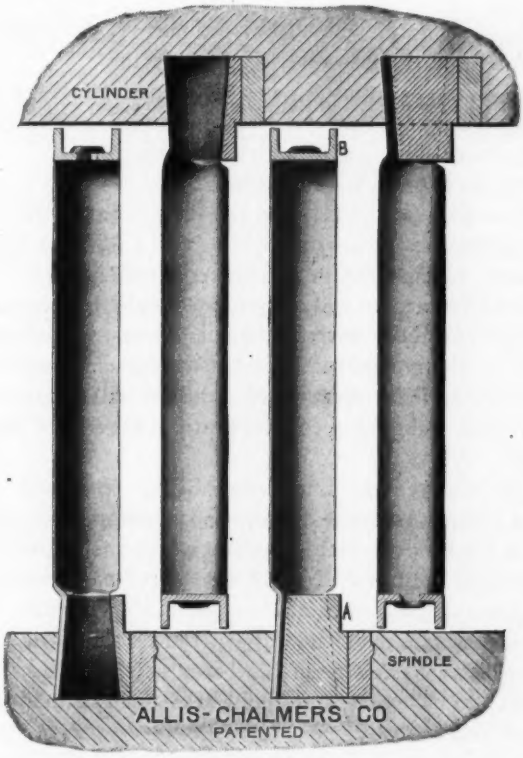
Plate IV.

maneuvering purposes, but such an arrangement would complicate the main piping as well as increase the weight, and, as far as known, has not been planned, except in the arrangement of H. M. *Dreadnought*, which has a backing turbine on each shaft.

The cruising turbines, usually consisting of separate high- and intermediate-pressure turbines, are attached to the outboard shafts when three shafts are used, and where most convenient, with regard to the arrangement, when with four shafts. They are only used in ships that speed at reduced powers for a considerable time when in service, as, for instance, naval vessels. The internal arrangements of these turbines are such as to admit an amount of steam corresponding to the power needed at the lower speeds only. The H.P. cruising turbine may thus be made for one-eighth the total power, while the I.P., when steam is led to it direct, may be proportioned for from one-third to one-half the power. They are given, comparatively speaking, a large diameter in order to get a suitable vane speed at the lower revolutions for which they must be designed. The steam passages between the vanes are also designed to correspond with the reduced steam quantity called for by the power when running slow. Besides the arrangements mentioned, the main turbines are sometimes arranged in units composed of a high-pressure, an intermediate-pressure and a low-pressure turbine—example, the *Tarantula*. With such an arrangement the L.P. turbine is placed on the center shaft and contains the backing turbine. This arrangement, while possessing some advantages, is deficient for maneuvering purposes, in that only the center screw is available for backing.

Construction.—A marine turbine consists essentially of parts included in the following enumeration, reference being had to Plates I, II, III and IV, which illustrate Parsons turbines designed for a recent installation of 16,500 horsepower. In this design is embodied the improved blading attachment as made in this country by the Allis-Chalmers Company, of Milwaukee, a detail of which is shown in Plate V. These

views do not represent the turbines chosen in the following examples, which are for a three-turbine arrangement of very much less power.



DETAIL OF BLADING.

Plate V.

1. The cylinder, which is usually divided in a lower casing (A) and an upper casing (B), bolted together at the center-line flanges by heavy bolts.

2. The cylinder heads (C) and (D), containing the stuffing boxes and bearing pedestals. These are sometimes separated from the heads, especially in very large turbines.

3. The rotor, which is connected to the turbine shafts (E) and (F), contains the ahead and backing drums (G) and (H). The backing drum is invariably put in the L.P. turbines only.

4. The blading, of which the stationary (J) is fastened to the cylinder and the movable (K) to the rotor.

5. The dummy pistons (L) and (P), composed of a series of thin rings fastened to the cylinder and running without contact in corresponding grooves turned in the rotor.

6. The steam-packed glands (M).

7. The main bearings (N) and the thrust bearing (O) in the forward pedestal.

8. Steam, exhaust, drain and oil connections.

While extreme care and judgment should be exercised in the design of steam turbines, great accuracy is absolutely essential in their construction. Only high-class methods of manufacture and the most careful design will insure to the turbine a place among recognized prime movers of standard efficiency.

Design.—A general knowledge being presumed of the essential features embodied in the construction of the turbine as well as the fundamental laws upon which the formulæ, used for its design, are based, it is not considered necessary here to enter upon a discussion of matters pertaining thereto. For a deeper investigation of the subject the reader is referred to such works as "The Steam Turbine," by Stodola; "Steam Turbines," by Thomas, and the works on Thermodynamics by Wood and Peabody.

In an attempt to design a steam turbine it is of the greatest importance for the designer to have knowledge of the degree of efficiency with which his machine is expected to operate. How to best secure this efficiency was not known in the earlier stage of the work, but was arrived at by costly experiments and repeated trials, which all involved the manufacturer in large expenditures, and therefore kept by him well guarded. No presumption of knowledge of whatsoever kind as to this is intended to form the basis of the following discussion, neither

is it possible to give data as to special features of constructions, knowledge of which has been gained by observation under privileged circumstances.

Principal attention will be directed to dimensions of blading as constituting the most important feature in the design, and will be treated for an arrangement of three shafts, consequently three turbines.

In order to determine the working efficiency of the system, we will assume losses according to the following table:

	In L.P. turbines. *Per cent.*	In H.P. turbine. *Per cent.*
Leakage over vane tips and in dummies, .	8	10
Mean steam friction in vanes in the "expansions,"	23	16
Energy loss due to velocity of exhaust, .	5	3
Fluid friction on rotating surfaces, mechanical friction in bearings and glands, etc., losses from radiation, etc., . . .	9	9
	45	38

Thermodynamic efficiency combined for all =

$$\frac{2\,(1 - .45) + 1\,(1 - .38)}{3} = 57.3 \text{ per cent.},$$

which means that the $\frac{\text{theoretical steam consumption}}{\text{actual steam consumption}} = .573.$

On account of the division of the total power among several turbines it will be necessary to determine, for each turbine, the temperature range within which it must operate, in order to enable a settlement as to number of "expansions," and the number of vane rows each "expansion" must contain, to develop the power assigned to the turbine on a certain quantity of steam predetermined by conditions.

By the term "expansion," used in the following, is meant a series of identical vane rows, both stationary and movable. The total vane system in each turbine is divided in a number

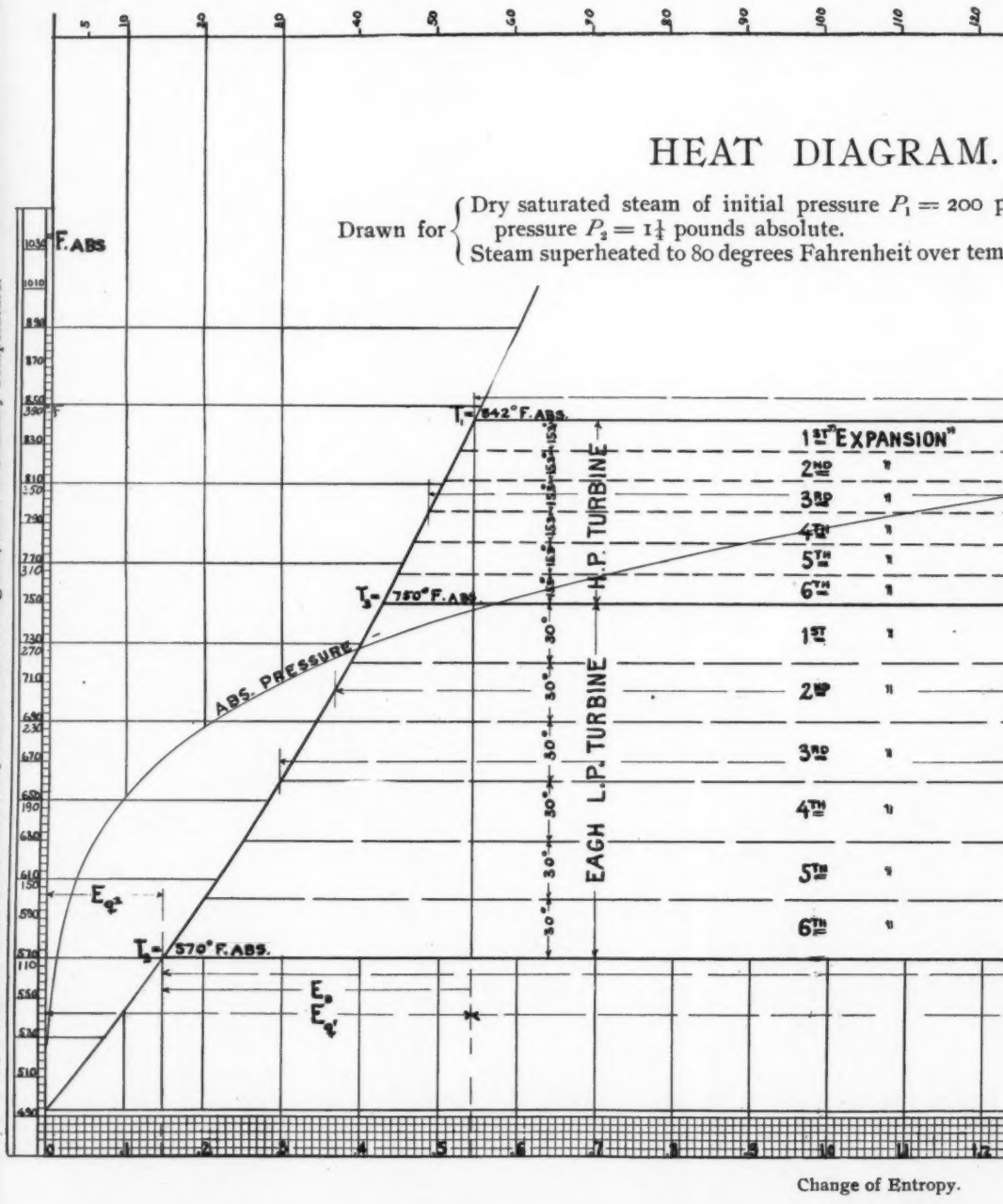
of such "expansions" or groups of vanes, and is done to facilitate both design and manufacture.

In three-turbine arrangements, with each developing one-third the power, and arranged as previously described, the high-pressure turbine, through which passes the whole steam quantity required for the total power, while each low-pressure turbine uses only one-half of this quantity, must be designed for a correspondingly lower range of working temperature, the limit of which is found by aid of the heat diagram illustrated in Figure 2. For our purpose the results will be sufficiently close, for the determination of the work equivalent to the total heat energy contained in a certain quantity of steam within a definite range of temperature, if the mean entropy of the temperature range in question be multiplied by the difference in temperature existing at entrance and exit of respective turbines by the steam quantity used. The heat in the steam available for work on the shaft is obtained by multiplying the total heat per pound by the thermodynamic efficiency, in some way established. Then, having reference to Figure 2, in which T_1 corresponds to the temperature at H.P. turbine inlet, T_2 ditto to L.P. outlet, T_3 to H.P. outlet and L.P. inlet, while T_3S divides the total heat area T_1KLT_2 in such parts that each represents an equal amount of work done per pound of the total steam used between the temperature range T_1, T_2, within which the steam expands, and observing the following designations:

E = mean entropy between T_1 and T_2;
T = number degrees Fahrenheit between T_1 and T_2;
Q' = total quantity of steam per second passing the turbine;
E' = mean entropy between T_1 and T_3;
T' = number degrees Fahrenheit between T_1 and T_3;
$\frac{Q'}{2}$ = half total quantity of steam per second;
E'' = mean entropy between T_3 and T_2;
T'' = number of degrees Fahrenheit between T_3 and T_2:

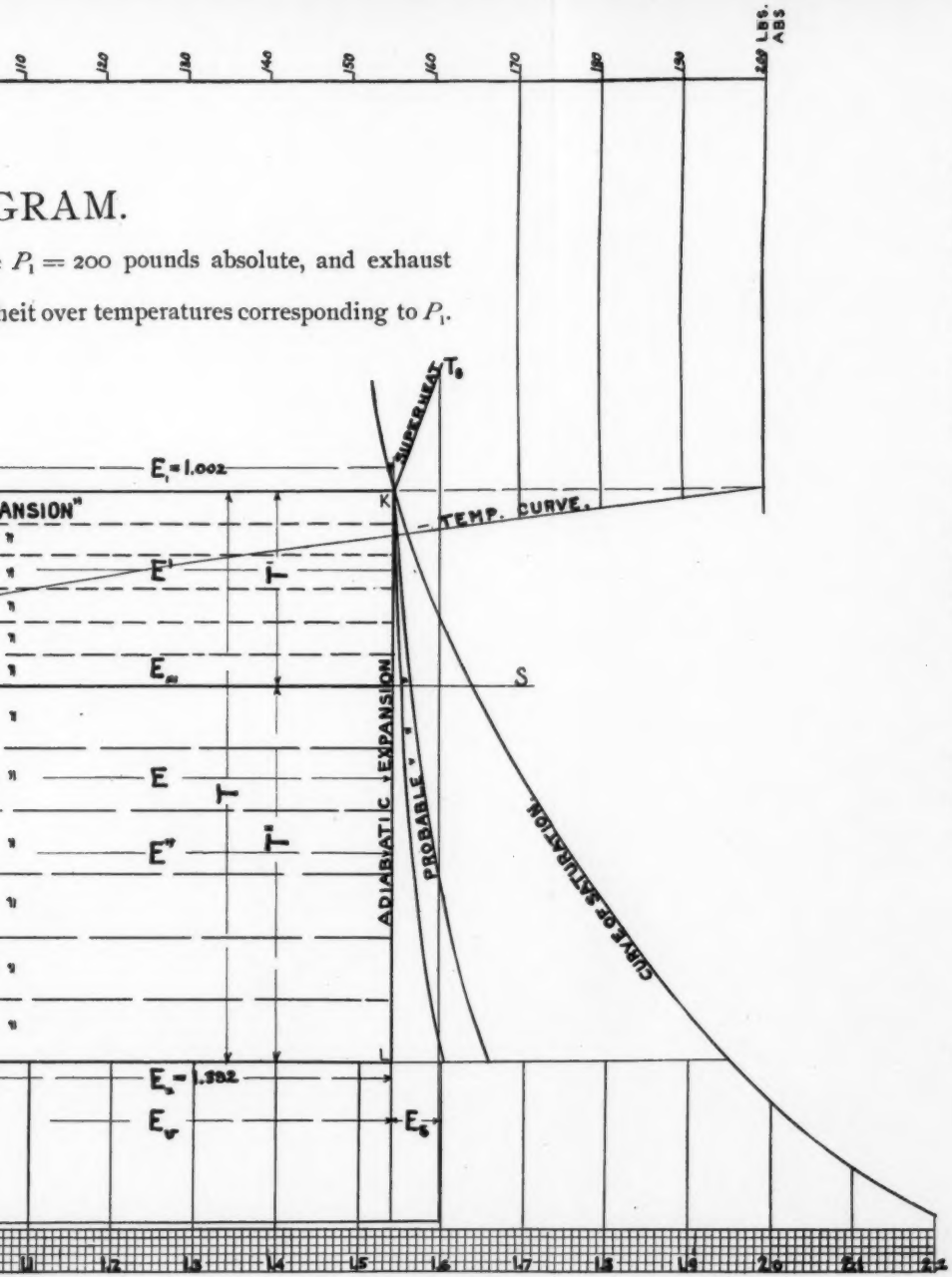
we may write this equation:

XU



Upper Figures, Scale of Absolute Temperature. Lower Figures, Scale of Ordinary Temperature.

Fig. 2.



f Entropy.

g. 2.

T_1 T_2 and
which
Q_1 = heat
Q_2 = heat
$H_v = E_v$
S_w = spec
S_s = spec
E_{q_1}, E_{q_2},
S_s (T_s —
$E_w = S_w$
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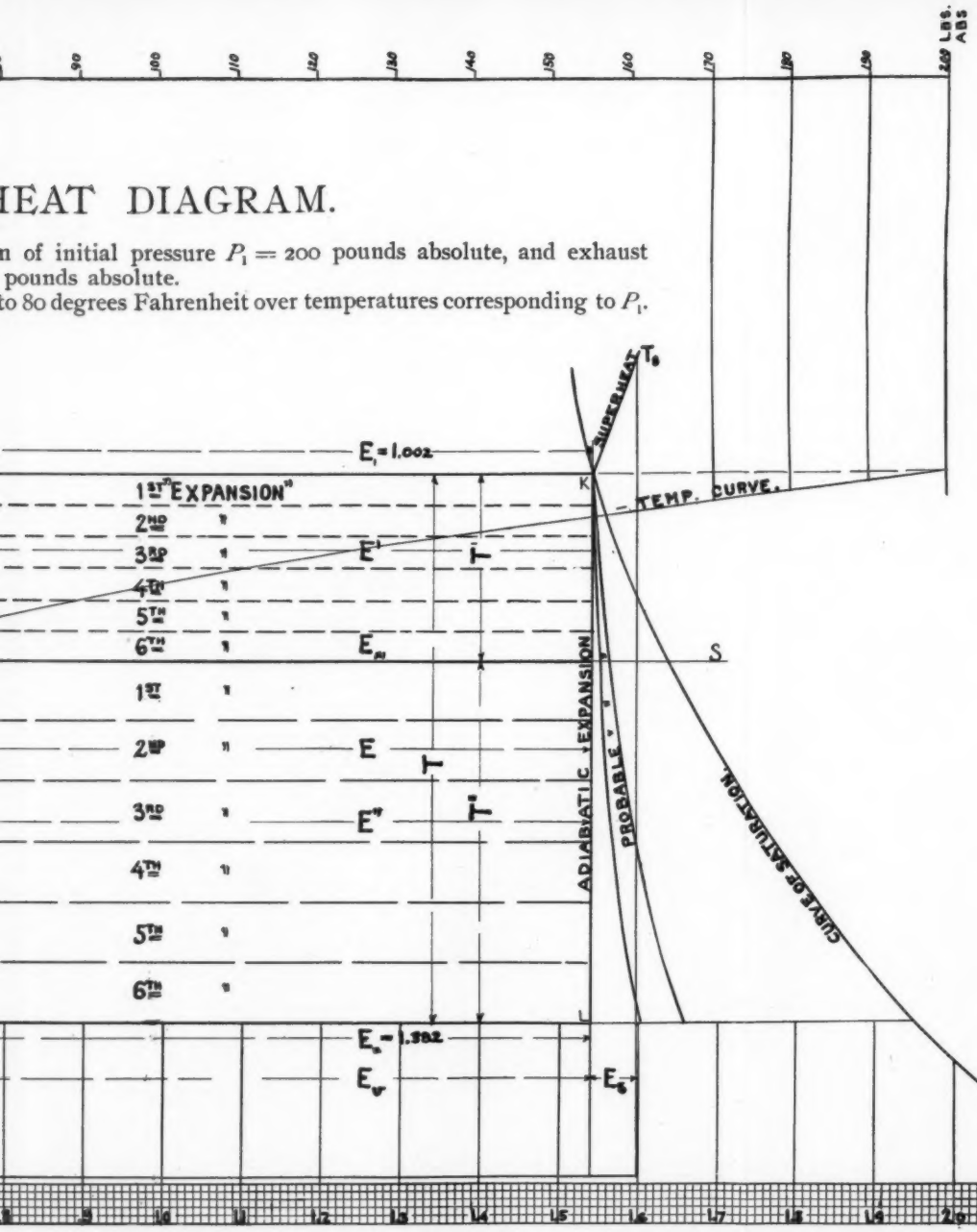
HEAT DIAGRAM.

n of initial pressure $P_1 = 200$ pounds absolute, and exhaust
pounds absolute.
to 80 degrees Fahrenheit over temperatures corresponding to P_1.



Change of Entropy.

Fig. 2.

EXPLANATION.

T_1 T_2 and K_1 T_s represent curves of change of entropy, for equations of which see below (1) and (2).

Q_1 = heat in liquid between T_1 and absolute zero.

Q_2 = heat in liquid between T_2 and absolute zero.

$H_v = E_v \times T_1$ = heat of vaporization. (See steam tables.)

S_w = specific heat of water.

S_s = specific heat of superheated steam.

E_{q_1}, E_{q_2}, E_v and E_s = change of entropy.

$S_s\ (T_s - T_1)$ = heat added by superheating from T_1 to T_s.

$$E_w = S_w \log e \frac{T_1}{T_2} \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (1)$$

$$E_s = S_s \log e \frac{T_s}{T_1} \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (2)$$

$(E_{q_1} + E_v - E_{q_2}) \times T_2$ = heat rejected in exhaust, saturated steam.

$(E_{q_1} + E_v - E_{q_2} + E_s) \times T_2$ = heat rejected in exhaust, superheated steam.

Change of entropy corresponding to H_v at any point $= \frac{H_v}{\text{F.° abs.}}$

H_1 = total heat in dry saturated steam per pound at P_1.

H_2 = total heat in dry saturated steam per pound at P_2.

H'_1 = total heat in superheated steam per pound at P_1.

H'_2 = total heat in superheated steam per pound at P_2.

Total heat per pound of dry saturated steam (not considering losses), for kinetic energy, $= Q_1 - Q_2 + H_v - T_2 (E_{q_1} + E_v - E_{q_2}) = H_1 - H_2$.

Total heat per pound of superheated steam (not considering losses), for kinetic energy =

$$Q_1 - Q_2 + H_v + S_s - T_2 (E_{q_1} + E^v - E_{q_2} + S_s \log e \frac{T_s}{T_1}) = H'_1 - H'_2.$$

Multiply total heat $H_1 - H_2$ and $H'_1 - H'_2$ by E_T to find available heat for kinetic energy.

Example with dry saturated steam (used in test):

$T_1 = 842$ degrees Fahrenheit, corresp. to $P_1 = 200$ pounds abs.

$T_2 = 570$ degrees Fahrenheit, corresp. to $P_2 = 1\frac{1}{4}$ pounds abs.

$Q_1 - Q_2$ (from steam tables) $= 354.9 - 78 = 276.9$ B.T.U.

$+ H_v = 843.6$

$1120.5 = H_1.$

$E_{q_1} + E_v - E_{q_2} = 1.395$

$\times T_2 = 1.395 \times 570 = 795.15 = H_2.$

$H_1 - H = 325.35$ B.T.U.

Example with superheated steam:

With T_1, T_2, Q_1, Q_2, and H_v as before, $H_1 = 1120.5$ B.T.U.

$S\ (T_s - T_1) = .58\ (922 - 842) = H_s = 46.4$ B.T.U.

$H'_1 = H_1 + H_s = 1166.9$ B.T.U.

$$E_s = .58 \log e \frac{922}{842} = 0.05421$$

$E_{q_1} + E_v - E_{q_2} + E_s = 1.449$

$\times T_2 = 1.449 \times 570 = H'_2 = 826.05$ B.T.U.

$H'_1 - H'_2 = 340.85$ B.T.U.

$$(E' \times T' \times Q') + (E'' \times T'' \times \frac{Q'}{2}) \times 2 = E \times T \times Q', \ldots (1)$$

Also, provided there were no losses, $E' \times T' = \frac{1}{3} E \times T = \frac{1}{3}$ area $T_1 K L T_2$, which represents one-third the total heat energy per pound of steam expanding between T_1 and T_2. But, owing to the various losses and to the fact that the friction losses of the steam within the turbine increase with decreasing pressures and with increasing velocities, it will be necessary to apportion to each heat area on each side of line $T_3 S$, such part of the energy loss caused by friction as most likely occurs. We shall assume, as before, an average steam-friction loss of 16 per cent. for the H.P. turbine, and 23 per cent. for each L.P. arrived at by experiments, also other losses, when on the basis of 57·3 per cent. combined efficiency, we get:

$$E' \times T' (1 - .38) = \frac{.573 \times E \times T}{3}; \; E' \times T' = .307 \, E \times T, \ldots (2)$$

and

$$(E'' \times T'') (1 - .45) = \frac{2 \times .573 \times E \times T}{3}; \; E'' \times T'' = .693 E \times T. (3)$$

in which formulae E', T', E'' and T'' must be determined.

This can be done in trial by assuming different values for E' and E'', T' and T'' until the equations are satisfied, but more directly by the following method:

Call entropy respectively at T_2, T_3 and T_1 E_{II}, E_{III} and E_1, and, considering averages sufficiently close for our purposes, we get:

$$E' = \frac{E_1 + E_{III}}{2}$$

$$E'' = \frac{E_{II} + E_{III}}{2}$$

$$\left(\frac{E_1 + E_{III}}{2}\right) (T_1 - T_3) = .307 \, E \times T$$

$$\left(\frac{E_{II} + E_{III}}{2}\right) (T_3 - T_2) = .693 \, E \times T.$$

Solving T_3 in the last equation and substituting in the first we get an expression from which E_{III} may be solved. This expression becomes complicated and is not at all necessary to give, as in all turbine problems T_1 and T_2, as well as E_1 and

E_{III}, are always known. For sake of simplicity we will apply the former to an actual example, in which is assumed:

Initial pressure at H.P. turbine inlet $= P_1 = 200$ pounds absolute, and $T_1 = 842$ degrees Fahrenheit, absolute.

Terminal pressure at L.P. turbine outlet $= P_2 = 1\frac{1}{4}$ pounds absolute, and $T_2 = 570$ degrees Fahrenheit, absolute, for which $E \times T = 326$ B.T.U. = total heat in steam, per pound, between temperatures stipulated. Then, having reference to Figure 2,

$$\left(\frac{E_{III} + 1.002}{2}\right)(842 - T_3) = .307 \times 326 = 100.$$

and

$$\left(\frac{E_{III} + 1.392}{2}\right)(T_3 - 570) = .693 \times 326 = 226.$$

After substitution and reduction we get

$$\left\{\frac{1.002 + E_{III}}{2}\right\}\left\{272 - \frac{452}{1.392 - E_{III}}\right\} = 100$$

from which, disregarding small discrepancies,

$$(E_{III})^2 = \frac{352}{272} = 1.3$$

$$E_{III} = \sqrt{1.3} = 1.14$$

$$E' = \frac{1.14 + 1.002}{2} = 1.071$$

$$T_1 = \frac{100}{1.071} = 93.3 \text{ and } T_3 = 842° - 93.3° = 748.7 \text{ degrees.}$$

Pressure corresponding to this temperature from tables= 57.72 pounds, absolute.

Having determined the temperature range and the pressures at inlet and outlet in each turbine, we will proceed to settle diameter of rotor and dimensions of blading. The drum forming the rotor spindle in a marine turbine is usually made of equal diameter right through in each machine, the low pressures, for obvious reasons, being much larger than the high pressure. With reference to Figure 5,

V_R = entering steam velocity.

V_S = leaving steam velocity due to expansion within each row of vanes.

V_V = mean vane speed due to speed of rotation.

Generally speaking, in all turbines V_S should have a fixed value in relation to the vane speed for maximum efficiency; in impulse-reaction turbines this occurs when $\frac{V_V}{V_S}$ is between $\frac{1}{2}$ and 1.

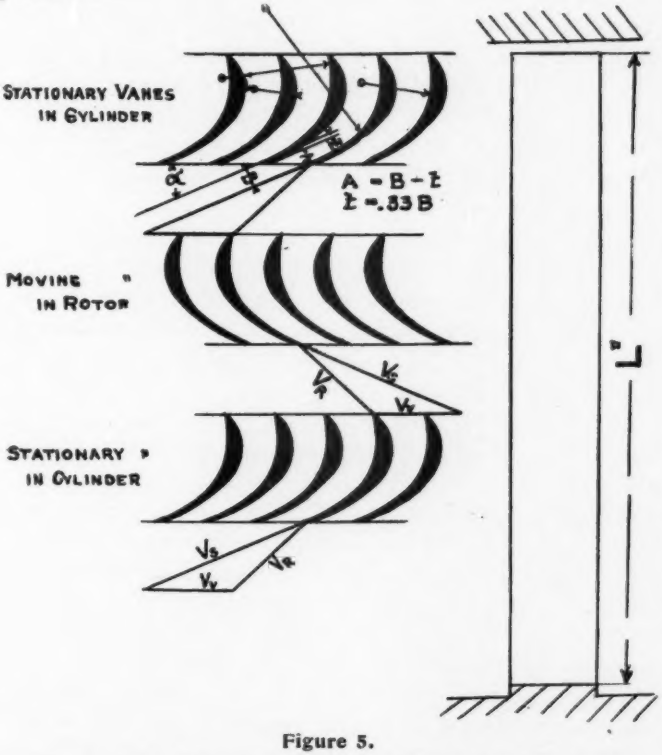


Figure 5.

DIAGRAMMATIC SKETCH SHOWING STEAM AND VANE VELOCITY.

In marine turbines it is not possible to attain for this ratio much above .4, and is oftener below, owing to the fact that its rotative speed is largely settled by the propeller and the diameter by other considerations. However, if kept between .3 and .4, an economy is assured which competes fairly well with present engines using about 13½ pounds steam per I.H.P.

A larger ratio could be obtained by decreasing V_S and keeping V_V constant, but in so doing, the heat drop in each row would also be decreased, which, in the turbine system, is equivalent to a proportionately greater number of rows of vanes and, therefore, a larger and heavier as well as a costlier turbine.

As stated in the beginning of this article, approximate outside dimensions of the turbine are arrived at with consideration to weight and space and approximate steam speed. To illustrate our proposition we will assume that each L.P. turbine must be limited to six feet in diameter and the H.P. to four feet ten inches and the revolutions not to exceed 650, which we found permissible with the propeller.

To enable us to determine the vane speed corresponding to the revolutions we must find the mean diameter of the vanes. Working inwards, making proper allowances for flanges, thickness of cylinder, clearance, etc., and knowing the steam quantity to be passed through, and approximating volumes for initial and final pressures and heat-units absorbed for certain increases of kinetic energy delivered corresponding to steam speeds of suitable relation to the vane speed, a preliminary value for vane lengths and angles may be obtained by successive trials for the first and last row. When mean vane diameter can be determined without restrictions as to outside dimensions this procedure is, of course, not necessary. For the problem in hand we get,

In L.P. the mean vane diameter = 4 feet 9 inches and $4\frac{1}{2}$-inch height of vane, with 45 degrees angle.

In H.P. the mean vane diameter = 3 feet $7\frac{1}{2}$ inches and $\frac{3}{4}$-inch height of vane, with 22 degrees angle.

The exit areas between the vanes in a steam turbine should, theoretically, increase in the same ratio as actual volumes, and thus along a curve resembling the pressure-volume curve. This is not done, however, for practical reasons, as it would involve the manufacturer in using too great a complexity of parts in the construction of the vane system. Neither is it at all necessary, practice having shown that good results accrue without. It is for this reason that divisions of the vanes are

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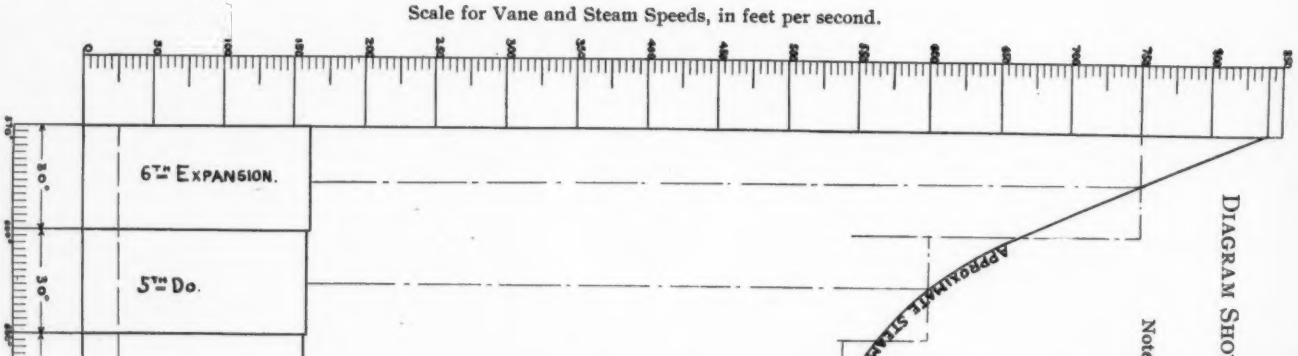
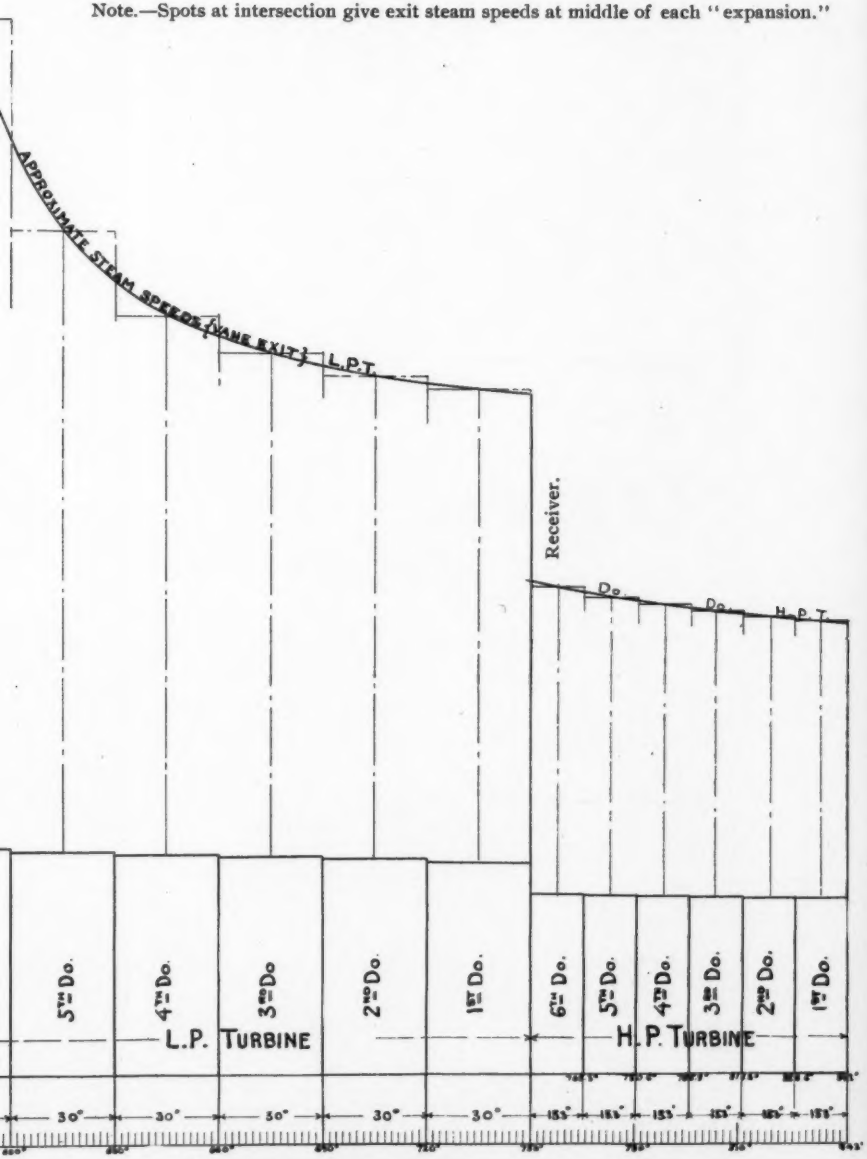


DIAGRAM SHOWING

Note

AGRAM SHOWING VANE AND STEAM SPEEDS, THROUGH THE TURBINE SYSTEM.

Note.—Spots at intersection give exit steam speeds at middle of each "expansion."



Scale for Range of Temperatures.

Fig. 3.

made in such a way that whole groups of vanes, commonly called "expansions" (see Figure 3), are made of exactly the same dimensions, the changes occurring only at reasonable intervals within each turbine. Each "expansion" is thus made to correspond to a certain temperature range, usually alike for all, while the same number of vanes are placed within each and all of the "expansions." In the majority of the "expansions" only the length of vane is increased, while in others, especially at the low-pressure end, where the volume increases very rapidly, for even small pressure differences, it also becomes necessary to change and enlarge the vane angle. It would be an almost endless task to attempt a formation of areas of the exit passages to correspond with theoretical conditions; neither is this done. The exit passage between the vanes of each "expansion" is, therefore, proportioned from average conditions existing within with reference to pressure as well as to quality and the volume of the steam, the weight of fluid passing remaining constant. Neither steam speeds nor the relation between vane and steam speed are constant—the former increase gradually towards the low-pressure end while the latter decreases, and the work performed per pound of steam throughout the turbine, in consequence thereof, is also variable.

Determination of Blading.—The exit velocity of the steam through the vanes within an "expansion" may be obtained from

$$V_S = \frac{Q \times S_M}{a}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

in which

Q = quantity, in pounds, passing the turbine per second;
S_M = mean specific volume, in cubic feet;
a = exit area, in square feet, between vanes in any one row;
V_S = steam speed, in feet, per second. (See Figure 5.)

The exit area is dependent on length L, in inches, of vane, the angle of exit α, thickness of vane t, and mean vane circumference C, in feet. Then by making the vane of such

dimensions that t becomes equal to one third the full opening between two vanes from face to face,

$$a = .66 \times \text{Sin} \propto \times \frac{L}{12} \times C.$$

The mean specific volume in each "expansion" is figured for average quality of the steam within as near as this can be determined. The quality is due to both the condition after its adiabatic expansion and the change that takes place during its flow as a result of friction between the vanes and the steam itself, causing heat and consequent re-evaporation, and is

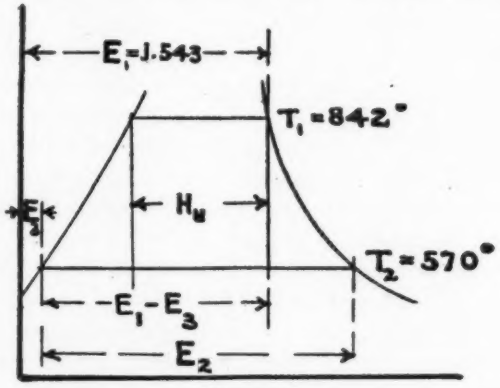


Figure 6.

equal to the sum of those quantities. The former may be gotten direct from the heat diagram by dividing entropy $E_1 - E_3$ at the adiabatic line with entropy at the saturation curve E_2 (see Figure 6), and the latter from equation

$$X = \frac{y\,(H_1 - H_2)}{H_v};$$

in which

y = percentage loss of kinetic energy due to friction;
H_1 = total heat of steam per pound at entrance of an "expansion";
H_2 = total heat rejected at the terminal of an "expansion";
H_v = heat of vaporization at pressure corresponding to H_2.

Example:

For expansion, within a range of temperature between 842 degrees and 570 degrees and $y = 30$ per cent., we get

$$\text{quality} = \frac{.30(1198.5 - 874.8)}{1037.5} + \frac{1.398}{1.82} = 86.2 \text{ per cent.},$$

which means that, of the originally dry steam of 100 per cent. quality, after expansion remains only 86.2 per cent. as steam, which is used for steam volume in determining the exit area, the water occupying a very insignificant volume.

The quantity or steam weight passing the turbines per second depends upon total horsepower as well the quantity needed per horsepower per hour, and may be taken, for varying conditions, from Table I. Owing to the fact that a large percentage of the total steam weight used, leaks through clearance spaces, both in vanes and dummy pistons, and does not pass through the vanes, an allowance, corresponding to the amount so wasted (assumed 8 per cent. in L.P. and 10 per cent. in H.P.), must be observed in figuring vane dimensions.

The problem now narrows down to figuring and designing the vane passages, both with regard to number and size, within each "expansion", for such velocities as, for their creation, will utilize all the available heat contained in the steam at stipulated initial and terminal pressures.

If steam enters a row of vanes with a velocity V_R and, on account of expansion within, attains velocity V_S at exit, the heat given up by the steam, in creating this condition, is found from the following fundamental formulæ:

$$\frac{V_S^2 - V_R^2}{2g} = (\lambda_1 - \lambda_2)\, 778, \quad . \quad . \quad . \quad . \quad (5)$$

in which

$\lambda_1 - \lambda_2$ = heat units given up by creating the increase in kinetic energy;

g = acceleration, in feet, per second = 32.16.

By assuming length and angle of vane, and having knowledge of the vane speed from speed of rotation and mean diameter of vanes from approximate outlay, V_S may be deter-

mined by use of formulæ (4) and V_R from the triangle of steam speeds illustrated for an actual example in Figure 4.

The increase in kinetic energy represented by $(V_S^2 - V_R^2) : 2g$ is never obtained, however, a large amount being lost on account of decreased velocities due to friction and re-heating of the steam; its value, therefore, must be diminished by an amount corresponding to the thermodynamic loss caused by the steam friction incidental upon conditions in a particular "expansion."

If K_E represents the mean efficiency of $\frac{V_S^2 - V_R^2}{2g}$ incidental to steam friction supposed to exist at each stage of the system, the kinetic energy delivered per pound of steam in foot pounds per second per row of vanes will be

$$= E_G = \frac{V_S^2 - V_R^2}{2g} \times K_E, \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Heat units absorbed per row in creating kinetic energy $= \frac{E_G}{778}$.

Furthermore, calling temperature range of each "expansion" T_R and the total temperature range T, corresponding to initial and final pressures at which each turbine operates, and assuming number of "expansions" $= n$, we get

$$T_R = \frac{T}{n}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

The available heat in B.T.U. per pound of steam for work at the coupling on the turbine shaft in each "expansion" from

$$T_R \times E_M \times T_E, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

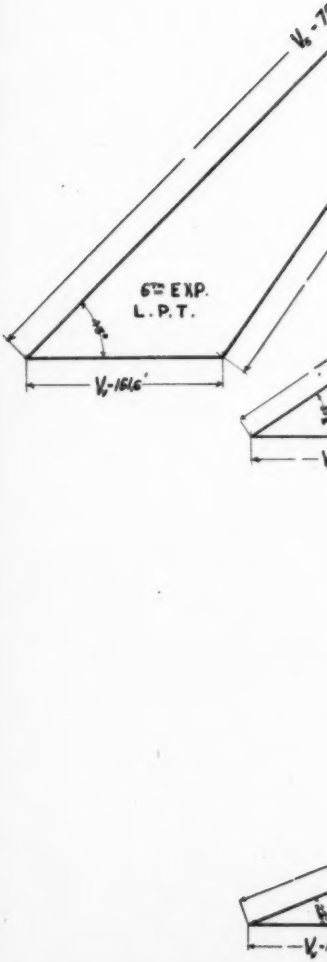
in which

E_M = mean entropy within the "expansion" corresponding to its range of temperature;

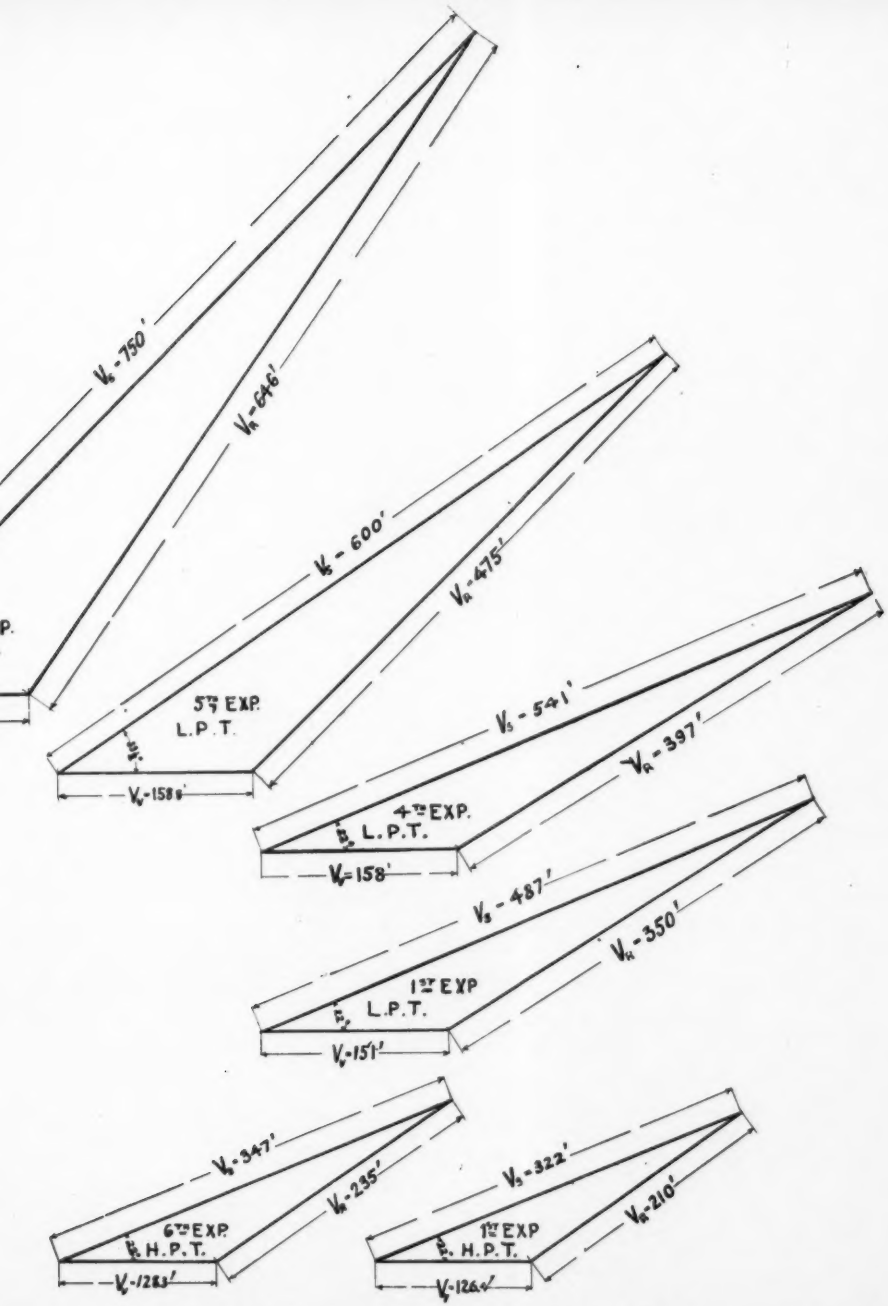
T_E = mean thermodynamic efficiency within the "expansion."

Number of rows of vanes in each "expansion"

$$= \frac{T_R \times E_M \times T_E \times 778}{Eg}, \quad . \quad . \quad . \quad . \quad . \quad (9)$$



SPEED TRIANGLES OF ENTR



LES OF ENTRANCE AND EXIT STEAM VELOCITIES WITH GIVEN VANE SPEEDS.

Fig. 4.

Table I.

STEAM CONSUMPTION, USING DRY SATURATED STEAM AT AND BETWEEN [illegible] POUNDS AND [illegible] POUNDS

Table I.

STEAM CONSUMPTION, USING DRY SATURATED STEAM AT AND BETWEEN 265 POUNDS AND 135 POUNDS INITIAL, AND 1 POUND AND 1¼ POUNDS TERMINAL, PRESSURES.

Thermo-dynamic efficiency = Er.

Absolute initial pressure, pounds per square inch. P_1	Absolute temperature in degrees Fahrenheit. T_1	Absolute terminal pressure, pounds per square inch. P_2	Absolute temperature in degrees Fahrenheit. T_2	Total heat in liquid. Q_1-Q_2	Heat of vaporization. H_v	Entropy. $E_1+E_v-E_2$	Heat lost in exhaust. $T_2\times(Eq_1+E_v-Eq_2)$	Total heat in steam per pound.	Available heat, in B.T.U. per pound, for kinetic energy.	Steam consumption in pounds per H.P. per hour = Q when	Steam consumption in pounds per H.P. per hour = Q when
...	$T_1=460+t_1$	.....	$T_2=460+t_2$	$H_v+Q_1-Q_2=H_1$		.....	$=H_2$	H_1-H_2	$E_T(H_1-H_2)=H$	$E_T=.55$	$E_T=.57$
265	866	1.0004	562	310.0	825.8	1.3951	784.0	351.8	193.5	13.15	12.69
265	866	1.2663	570	302.0	825.8	1.3792	786.0	341.8	188.0	13.54	13.06
265	866	1.5039	576	296.0	825.8	1.3687	788.0	333.8	183.6	13.86	13.4
265	866	1.7789	582	290.0	825.8	1.3583	790.0	325.8	179.2	14.2	13.7
200	842	1.0004	562	284.9	843.6	1.4103	792.6	335.9	E_T assumed = 0.55. 184.2	13.8	13.32
200	842	1.2663	570	276.9	843.6	1.3944	794.8	325.7	179.2	14.2	13.7
200	842	1.5039	576	270.9	843.6	1.3839	797.1	317.4	174.6	14.57	14.06
200	842	1.7789	582	264.8	843.6	1.3735	799.4	309.0	170.0	14.97	14.45
135	810	1.0004	562	251.4	867.3	1.4423	810.6	308.1	169.4	15.2	14.67
135	810	1.2663	570	245.4	867.3	1.4264	813.2	299.5	164.7	15.45	14.91
135	810	1.5039	576	237.4	867.3	1.4159	815.6	289.1	159.0	16.00	15.44
135	810	1.7789	582	231.3	867.3	1.4055	818.0	280.6	154.3	16.49	15.91

t_1 = temperature of steam at P_1.

t_2 = temperature of steam at P_2.

For reference, see figure 6.

$$Q = \frac{60 \times 33{,}000}{(H_1 - H_2)\ 778 \times E_T} = \frac{2{,}545}{H}$$

$$E_T = \frac{H}{H_1 - H_2}$$

Horsepower transmitted to shaft by each "expansion"

$$= Q \times T_R \times E_M \times T_E \times \frac{778}{550}, \quad . \quad . \quad . \quad . \quad (10)$$

in which Q is steam, in pounds, passing turbine per second.

The use of foregoing formulæ may be illustrated best by application to an actual case in which we may assume the following particulars, which are substantially the same as have already been used.

Total brake horsepower = 6,500;

An arrangement of three turbines of equal power.

Thermodynamic efficiency = 57 per cent.

Steam consumption from Table I = 13.7 pounds.

Total steam used per second = 25 pounds.

Total steam passing vanes in H.P. turbine, $.9 \times 25 = 22.5$ pounds.

Total steam passing vanes in L.P. turbine, $\frac{.92 \times 25}{2} = 11.5$ pounds.

Number of "expansions" in each turbine = 6.

Initial pressure at H.P. = 200 pounds absolute.

Terminal pressure at L.P. = $1\frac{1}{4}$ pounds absolute.

R. P. M. = 650.

Mean vane diameter last row L.P. = 4 feet 9 inches.

Mean vane diameter first row H.P. = 3 feet $7\frac{1}{2}$ inches.

The figuring, being substantially the same for any one "expansion," will be executed for only a few of these in each turbine.

Figure 2 represents the heat diagram corresponding to the example chosen. Having found T_3, with T_1 and T_2 given, line $T_3 S$ forms the dividing line, in the "total heat area," at which the H.P. and L.P. turbines are to work. The areas above and below this line are then subdivided into equal parts, the upper by differences of $\frac{93.3}{6} = 15.3$ degrees and the lower by $\frac{178.7}{6}$ = approximately 30 degrees. Each division represents an "expansion."

The combined mean quality of the steam is determined within each of said "expansions" from quality due to conditions before and after "expansion" (see Table II).

Mean steam quantities, in cubic feet, are determined by multiplying the mean specific volume by the mean quality (see Table III). In figuring the steam speeds a considerable amount of compromise must be resorted to, first, because of the fact that it is necessary to see to it that the ratio $\frac{V_V}{V_S}$ comes within limits such as will give an efficiency to correspond with the stipulated steam consumption; secondly, because of the definite relation steam speeds bear to the kinetic energy increase and heat absorption.

The number of vane rows in each "expansion" must be sufficient to absorb its available heat units, and the total number of "expansions" in each turbine large enough to absorb all the available heat between pressures determined. Tabulating results we get for

L.P. TURBINE.

6th "expansion."

0. Total steam flow in pounds per second through each L.P. turbine $= \frac{25}{2}$, leakage amounting to 8 per cent. leaves 11.5 pounds flowing through vanes.
1. Range of temperature, between 570 degrees and 600 degrees = 30 degrees = T_R.
2. Mean specific volume = 193.5 cubic feet = S_M.
3. Mean entropy = 1.373 = E_M.
4. Mean quality = .87 per cent.
5. Mean quantity = $.87 \times 195.3 = 170$ cubic feet = $_S$.
6. Mean efficiency of $\frac{V_S^2 - V_R^2}{2g}$ observing steam friction $\left\{\begin{matrix} 1 - .30 \\ 1 - .27 \end{matrix}\right\} : 2 = 71.5$ per cent. = K_E.

Table II.—ENTROPY AND QUALITY OF STEAM.

Quality due to adiabatic expansion and re-evaporation by friction.

Entropy.				Quality due to adiabatic expansion.	Quality due to re-evaporation.	Combined quality $=Q$.
Abs. temp.	E_3	E_2	$E_1=.5432+\frac{843.6}{842}=1.5452$ E_1-E_3	$X_1=\frac{E_1-E_2}{E_2}$	$X_2=y\frac{(H_1-H_2)}{H_v}$	$Q=X_1+X_2$
570	$=.147$	$=\frac{1037.5}{570}=1.82$	$1.545-.147=1.398$	$1.398\div1.82=.768$	$=\frac{.30(1,198.5-874.8)}{1,037.5}=.093$	$.768+.093=.862$
600	$=.198$	$=\frac{1016}{600}=1.693$	$1.545-.198=1.347$	$1.347\div1.693=.790$	$=\frac{.27(1,198.5-863.5)}{1,016.4}=.088$	$.795+.089=.878$
630	$=.247$	$=\frac{995.3}{630}=1.58$	$1.545-.247=1.298$	$1.298\div1.58=.821$	$=\frac{.25(1,198.5-955)}{995.3}=.061$	$.821+.063=.884$
660	$=.294$	$=\frac{974.2}{660}=1.476$	$1.545-.294=1.251$	$1.251\div1.476=.847$	$=\frac{.23(1,198.5-994)}{974.2}=.048$	$.847+.048=.895$
690	$=.339$	$=\frac{953.2}{690}=1.381$	$1.545-.339=1.206$	$1.206\div1.381=.873$	$=\frac{.21(1,198.5-1,036)}{953.2}=.036$	$.873+.036=.908$
720	$=.382$	$=\frac{932}{720}-1.294$	$1.545-.382=1.163$	$1.163\div1.294=.893$	$=\frac{.19(1,198.5-1,069)}{932}=.026$	$.893+.026=.919$
750	$=.423$	$=\frac{911}{750}=1.215$	$1.545-.423=1.122$	$1.122\div1.215=.923$	$=\frac{.17(1,198.5-1,100)}{911}=.018$	$.925+.018=.940$
765.3	$=.443$	$=\frac{900}{765.3}=1.175$	$1.545-.523=1.022$	$1.102\div1.175=.938$	$=\frac{.17(1,198.5-1,118.5)}{900}=.015$	$.938+.015=.953$
842	$=.543$	$=\frac{843.6}{842}=1.002$	$1.545-.543=1.002$	$1.002\div1.002=1.00$	$=\frac{.15(1,198.5-1,198.5)}{843.6}=.00$	$.100+0.0=1.00$

H_1 = total heat per pound between T_1 and absolute 0 degrees = 1,198.5, from tables.
H_2 = total heat per pound rejected at T_2 = 874.8, from $H_2 = X_1 \times H_v + Q_2$.
H_v = heat of vaporization at H_2 = 1,037.5,, from tables.
No allowance made for drop in receiver.

Table III.

MEANS OF SPECIFIC VOLUMES, ENTROPY, QUALITY AND QUANTITY.

L.P. TURBINE

	Range of temperature = 30 degrees Fahrenheit.	Mean specific volume.	Mean entropy.	Mean quality.	Mean quantity.
	degrees.				*cubic ft.*
6th expansion.	570 600	267.5 123.2 } : 2 = 195.3	570 = 1.398 600 = 1.347 } : 2 = 1.373	.862 .871 } : 2 = .87	170
5th expansion.	600 630	123.2 61.58 } : 2 = 92.39	600 = 1.347 630 = 1.298 } : 2 = 1.323	.871 .884 } : 2 = .881	81.71
4th expansion.	630 660	61.58 33.40 } : 2 = 47.5	630 = 1.298 660 = 1.251 } : 2 = 1.275	.884 .895 } : 2 = .89	42.3
3d expansion.	660 690	33.40 19.20 } : 2 = 26.30	660 = 1.251 690 = 1.206 } : 2 = 1.229	.895 .908 } : 2 = .901	23.67
2d expansion.	690 720	19.20 11.60 } : 2 = 15.40	690 = 1.206 720 = 1.163 } : 2 = 1.185	.908 .919 } : 2 = .913	14.2
1st expansion.	720 750	11.60 7.35 } : 2 = 9.5	720 = 1.163 750 = 1.122 } : 2 = 1.142	.919 .940 } : 2 = .93	8.84

H.P. TURBINE.

	Range of temperature = 15.3 degs. Fahrenheit.	Mean specific volume.	Mean entropy.	Mean quality.	Mean quantity.
6th expansion.	750 765.3	7.35 5.95 } : 2 = 6.65	1.122 1.098 } : 2 = 1.109	.94 .953 } : 2 = .946	6.3
5th expansion.	765.3 780.6	5.95 4.86 } : 2 = 5.4	1.098 1.075 } : 2 = 1.086	.975	5.26
4th expansion.	780.6 795.9	4.86 4.004 } : 2 = 4.43	1.075 1.058 } : 2 = 1.066	.978	4.36
3d expansion.	795.9 811.2	4.004 3.76 } : 2 = 3.882	1.058 1.038 } : 2 = 1.048	.985	3.84
2d expansion.	811.2 826.5	3.76 2.78 } : 2 = 3.27	1.038 1.02 } : 2 = 1.029	.99	3.24
1st expansion.	826.5 842	2.78 2.28 } : 2 = 2.53	1.02 1.002 } : 2 = 1.011	.99 1.00 } : 2 = .995	2.52

H_2 = heat of vaporization at H_2 = 1,037.5,, from tables.
No allowance made for drop in receiver.

7. Vane exit angle $\alpha = 45$ degrees.
8. Length of vane $= 4\frac{1}{2}$ inches.
9. Mean vane diameter $= 4$ feet 9 inches.
10. Mean vane circumference $= 14.92$ feet.
11. Effective exit area $= .66 \times \sin. 45° \times \frac{4\frac{1}{2}}{12} \times 14.92 = 2.605$ square feet.
12. Vane speed $V_V = \frac{650 \times 14.92}{60} = 161.6$ feet per second.
13. Steam speed $V_S = \frac{Q \times s}{a} = \frac{11.5 \times 170}{2.605} = 750$ feet per second, leaving.
14. Steam speed $V_R =$ from speed triangle $= 646$ feet per second, entering.
15. Kinetic energy delivered per pound of steam in foot pounds per second, per row of vanes $E_G = \frac{V_S^2 - V_R^2}{2g} \times K_E = 1{,}615$.
16. Heat units absorbed per row in creating kinetic energy $= \frac{1{,}615}{778} = 2.077$.
17. Available B.T.U. per pound of steam for work on shaft in 6th "expansion" $= T_R \times E_M \times T_E = 30 \times 1.373 \times .495 = 20.4$.

T_E, mean thermodynamic efficiency, is derived as follows:

at $\left\{\begin{array}{l} 570°, \text{ friction } 30 \text{ per cent., } T_E = 1 - .52 = .48 \\ 600°, \text{ friction } 27 \text{ per cent., } T_E = 1 - .49 = .51 \end{array}\right\} : 2 = 495.$

18. Number of rows of vanes in 6th "expansion" $= \frac{20.4}{2.077} = 10.$
19. Horsepower (total steam $= \frac{25}{2}$ lbs.) $= \frac{25}{2} \times \frac{20.4 \times 778}{550} = 361.$

5th "expansion."

1. Range of temperature between 600° and 630° $= T_R$.
2. Mean specific volume $= 92.39 = S_M$.
3. Mean entropy $= .1323 = E_M$.

4. Mean quality = .881 per cent.
5. Mean quantity = .881 × 92.39 = 81.71.
6. Mean $K_E \left\{ \begin{matrix} 1-.27 \\ 1-.25 \end{matrix} \right\} : 2 = .74$.
7. Vane exit angle $\alpha = 33\frac{1}{2}$ degrees.
8. Vane length = $3\frac{1}{2}$ inches.
9. Mean vane diameter = 4 feet 8 inches.
10. Mean vane circumference = 14.66 feet.
11. Effective exit area = $.66 \times \sin. 32\frac{1}{2}° \times \frac{3\frac{1}{2}}{12} \times 14.66 = 1.56$ square feet.
12. Vane speed $V_r = 158.8$ feet per second.
13. Steam speed $V_S = \frac{11.5 \times 81.71}{1.56} = 600$ ft. per second, leaving.
14. Steam speed V_R from triangle = 475 feet per second, entering.
15. $E_G = \frac{V_S^2 - V_R^2}{2g} \times K_E = \frac{360{,}000 - 226{,}576}{2 \times 32.16} \times .74 = 1{,}535.$
16. Heat units absorbed $= \frac{1{,}535}{778} = 1.974$ per row of vanes.
17. Available B.T.U. in steam in 5th "expansion" for work on shaft = $T_R \times E_M \times T_E = 30 \times 1.323 \times .52 = 20.63$.

T_E at $\left\{ \begin{matrix} 600° = 1-.49 \text{ with 27 per cent. friction} \\ 630° = 1-.47 \text{ with 25 per cent. friction} \end{matrix} \right\} : 2 = .52.$

18. Number of rows of vanes in 5th expansion $= \frac{20.63}{1.974} = 10.4$, and we will make this even 10 rows.
19. Horsepower $= \frac{25}{2} \times \frac{20.63 \times 778}{550} = 350.$

4th "expansion."

1. Temperature range between 630 and 660 degrees.
2. Mean specific volume = 42.3.
3. Mean entropy $E_M = 1.275$.

4. Mean quality = .89 per cent.
5. Mean quantity = 42.3.
6. Mean $K_E = \left\{ \begin{matrix} 1 - .25 \\ 1 - .23 \end{matrix} \right\} : 2 = .76.$
7. Vane angle α = 22 degrees.
8. Vane length = 3 inches.
9. Mean vane diameter = 4 feet $7\frac{1}{2}$ inches.
10. Mean vane circumference = 14.6 feet.
11. Effective exit area = .66 sin. 22 degrees $\times \frac{3}{12} \times$ 14.6 = .9024 square feet.
12. Vane speed V_V = 158 feet per second.
13. Steam speed $V_S = \frac{11.5 \times 42.3}{.9024} = 541$, leaving.
14. Steam speed V_R, from triangle = 397, entering.
15. $E_G = \frac{541^2 - 397^2}{64.32} \times .76 = 1594.$
16. Heat units absorbed $= \frac{1,594}{778} = 2.05.$
17. Available B.T.U. in steam in 4th expansion for work on shaft = 30 × 1.278 × .54 = 20.71.
18. Number of rows of vanes in 4th expansion $= \frac{20.71}{2.05} = 10.0.$
19. Horsepower $= \frac{25}{2} \times \frac{20.71 \times 778}{550} = 350.$

1st "expansion."

0. Total steam same as before.
1. Range of temperature between 720 degrees and 750 degrees = 30 degrees = T_R.
2. Mean specific volume = 9.5 cubic feet = S_M.
3. Mean entropy = 1.142 = E_M.
4. Mean quality = .96 per cent.
5. Mean quantity = 9.5 × .93 = 8.9 cubic feet.

6. Mean K_E from $\left\{\begin{matrix}1-.19\\1-.17\end{matrix}\right\}: 2 = 82$ per cent.
7. Vane exit angle $\alpha = 22$ degrees.
8. Length of vane $= \frac{3}{4}$ inch.
9. Mean vane diameter $= 4$ feet $5\frac{1}{4}$ inches.
10. Mean vane circumference $= 13.92$ feet.
11. Effective exit area $= .66 \times \sin. 22 \times \frac{\frac{3}{4}}{12} \times 13.92 = .21$ sq. ft.
12. Vane speed $V_V = \frac{650 \times 13.92}{60} = 151$ feet per second.
13. Steam speed $V_S = \frac{11.5 \times 8.9}{.21} = 487$ feet per second, leaving.
14. Steam speed V_R from triangle $= 350$ feet per second, entering.
15. Energy delivered $E_G = \frac{487^2 - 350^2}{64.32} \times .82 = 1462$ ft. lbs.
16. Heat units absorbed per row $= \frac{1462}{778} = 1.88$.
17. Available B.T.U. $= 30 \times 1.142 \times .60 = 20.54$.

$T_E = \left\{\begin{matrix}1 - .39 \text{ with friction} = 17 \text{ per ct.} = .61\\1 - .41 \text{ with friction} = 19 \text{ per ct.} = .59\end{matrix}\right\}: 2 = .60.$

18. Number of rows in 1st "expansion" $= \frac{20.54}{1.88} = 10.9$.
19. Horsepower 1st "expansion" $= \frac{20.54 \times 778}{550} \times \frac{25}{2} = 370$.

H.P. TURBINE.

6th "expansion."

0. Steam per second used $= 22.5$ pounds.
1. Range of temperature between 750 degrees and 765.3 degrees $= 15.3$ degrees $= T_R$.
2. Mean specific volume $= 6.65$ degrees $= S_M$.
3. Mean entropy $= 1.109 = E_M$.
4. Mean quality $= .946$ per cent.
5. Mean quantity $= 6.3$ cubic feet.

6. Mean $K_E = 83$ per cent. from $\left\{ \begin{matrix} 1 - .17 \\ 1 - .17 \end{matrix} \right\} : 2 = .83$.
7. Length of vane $= 1\frac{3}{4}$ inches.
8. Mean vane diameter $= 3$ feet $9\frac{1}{4}$ inches.
9. Mean vane circumference $= 11.85$ feet.
10. Vane exit angle $= 22$ degrees.
11. Effective exit area $= .66 \times \sin 22$ degrees $\times \frac{1\frac{3}{4}}{12} \times 11.85$ $= .42$ square feet.
12. Vane speed $V_v = \frac{650 \times 11.85}{60} = 128.3$ feet per second.
13. Steam speed $V_S = \frac{23 \times 6.65}{426} = 359$ feet per second, leaving.
14. Steam speed V_R from triangle $= 245$ feet per second, entering.
15. Energy delivered $E_G = \frac{359^2 - 245^2}{64.32} \times .83 = 889$ foot pounds.
16. Heat units absorbed per row $= \frac{889}{778} = 1.142$.
17. Available B.T.U. $= 15.3 \times 1.109 \times .63 = 11.68$.
18. Number of rows in 6th "expansion" $= \frac{11.68}{1.142} = 10.2$.
19. Horsepower $= \frac{11.68 \times 25 \times 778}{550} = 393$.

1st "expansion."

0. Steam per second used $= 22.5$ pounds.
1. Range of temperature between 826.5 degrees and 842 degrees $= 15.5$ degrees.
2. Mean specific volume $= 2.53$.
3. Mean entropy $= 1.01 = E_M$.
4. Mean quality $= .996$ per cent.
5. Mean quantity $= 2.52$.

6. Mean $K_E = .8475$.
7. Vane angle = 22 degrees.
8. Length of vane $\frac{3}{4}$ inch.
9. Mean vane diameter = 3 feet $8\frac{1}{2}$ inches.
10. Mean vane circumference = 11.67 feet.
11. Effective exit area $= .66 \times \sin. 22 \frac{\frac{3}{4}}{12} \times 11.67 = .18$ sq. ft.
12. Vane speed $V_V = \frac{650 \times 11.67}{60} = 124.6$ feet per second.
13. Steam speed $V_S = \frac{22.5 \times 2.52}{.18} = 322$ feet per second.
14. Steam speed V_R (from triangle) = 217 feet per second.
15. E_G, or energy given up $= \frac{322^2 - 217^2}{64.32} \times .847 = 747$.
16. Heat units absorbed per row $= \frac{747}{778} = .9604$.
17. Available B.T.U. $= 15.5 \times 1.01 \times .64 = 9.63$.

Number of rows of vanes in 1st "expansion" $= \frac{9.63}{9.604} = 10$.

Horsepower $= \frac{9.63 \times 778}{550} \times 25 = 342$.

The ratio vane to steam speed ranges in the above examples between .22 for the last low-pressure "expansion" and .39 for the first high-pressure. The average horsepower in each "expansion" is about 365, which for eighteen "expansions" of the three turbines gives a total of 6570.

The foregoing references to blading pertains only to such dimensions as length, exit angles and numbers of rows and "expansions," and does not treat other features of construction, such as entrance angles, clearances, spacing and material, all of which enter largely in securing maximum efficiency. Special attention must, of course, be paid to these details when designing with manufacturing in view, which is not within the scope of this paper.

UNBALANCED THRUST.

A feature of great practical consequence in marine turbines of Parsons type ensues from the fact that the unbalanced pressure on the vane system, to a large extent, counteracts the effective thrust of the propeller. The turbines are, therefore, always placed in the ship in such a manner that the steam flows in a direction opposite to the motion of the ship. The thrust caused by the propeller along the shaft may be counterbalanced to any degree desired by giving to the drum extension, forming the dummy piston, such dimensions that the end thrust on the rotor together with the unbalanced vane thrust will produce the proper distribution of pressure on the thrustblock. Owing to the fact that the greater part of the thrust required for driving the ship is transmitted through the after bearing and its holding-down bolts to the foundations, makes it possible to greatly reduce the dimensions of the thrustblock.

The unbalanced pressure on the rotor is caused by the successive drop in pressure between one row of vanes and the next following, and is found by multiplying the area of the vane annulus by the difference in the pressure existing in the clearance space on each side of the vanes. The axial velocity components need be taken into account only when the degree of reaction, or the work done by the steam in the moving vanes divided by the total work, is more or less than one-half; usually this is arranged to equal one-half, and is so arranged in previous examples.

The pressure in the clearance spaces corresponding to the temperatures will be found for each "expansion" direct from the steam tables, the area of the annulus from vane dimensions and the axial components from triangle of steam speeds. We then get in the ahead turbine as unbalanced pressure in each row,

$$T_a = A \times 144\,(p_1 - p_2) \pm \frac{w\,(V_k^1 - V_k)}{g},$$

in which

A = area of one vane annulus, in square feet.

p_1 = pressure in pounds per square inch in clearance space, forward side.

p_2 = pressure in pounds per square inch in clearance space, after side.

w = steam weight passing per second.

g = acceleration in pounds per second = 32.16.

V_k' and V^k = axial components of steam speed V_S. Applied to the sixth "expansion" of the L.P. turbine of our previous example we get temperature drops and pressures for each moving vane row as follows:

1st { 597 degrees Fahrenheit = 2.66 pounds.
594 degrees Fahrenheit = 2.46 pounds, difference = .2 pound per square inch.

2d { 591 degrees Fahrenheit = 2.27 pounds.
588 degrees Fahrenheit = 2.09 pounds, difference = .18 pound per square inch.

3d { 585 degrees Fahrenheit = 1.93 pounds.
582 degrees Fahrenheit = 1.77 pounds, difference = .16 pound per square inch.

4th { 579 degrees Fahrenheit = 1.63 pounds.
576 degrees Fahrenheit = 1.503 pounds, difference = .127 pound per square inch.

5th { 573 degrees Fahrenheit = 1.38 pounds.
570 degrees Fahrenheit = 1.26 pounds, difference = .12 pound per square inch.

Total equivalent on one ring = .787 pound per square inch.

Area of annulus $A \times 144 = 5.6 \times 144 =$ square inches.

Axial components being neutralized by design, T_u for sixth "expansion" $= 5.6 \times 144 \times .787 = 630$ pounds.

T_u for fifth "expansion" $= 4.2 \times 144 \times 1.5 = 900$ pounds.

T_u for first "expansion" $= .84 \times 144 \times 9 = 1088$ pounds.

Suppose that the unbalanced pressure in all of the six "expansions" equals 4,650 pounds. By previous figures the

effective thrust in the example chosen equals 15,400 pounds, then pressure of propeller over unbalanced vane thrust equals 15,400—4,650=10,750 pounds. Now by proportioning the rotor extension or dummy diameter in such a way that the end pressure, due to initial steam pressure of any particular turbine, will amount to say 6,600 pounds, the difference, or 4,150 pounds, will be transmitted through the thrustblock, and its dimensions may be determined by this pressure. The difference between effective thrust due to propeller, here 15,400 pounds, and the amount transmitted through the thrustblock gives the thrust which must be exerted through the turbine casing to drive the ship. Through the turbine casing all of the thrust is in fact transmitted, as the thrustblock is either cast solid with the casing or bolted to it.

It will be appreciated by engineers accustomed to the accuracy and certainty with which designs of reciprocating engines can be executed that, in making turbine designs, it is not possible to work as readily from given stipulations for direct results. These can be obtained, in starting, only by previous approximations and various assumptions. However, as the art progresses and turbine work becomes more general, tabulation of the quantities which principally influence dimensions, and by limiting conditions, especially with regard to initial and terminal pressures, dry saturated or superheated steam, revolutions and ratios of vane and steam speeds, will greatly facilitate an otherwise very laborious calculation.

PERFORMANCE OF THE ASSISTANT CYLINDERS OF THE *WASHINGTON.*

By Ensign W. W. Smith, U. S. N., Member.

The work of taking the indicator cards and analyzing the results was done by the officers under instruction in Steam Engineering under the direction of Lieutenant Commander B. C. Bryan, U. S. N.

Complete descriptions of the principles and working of the Lovekin Assistant Cylinder have been published in the Journal of the American Society of Naval Engineers, Vol. XVI, Nos. 1 and 3, Vol. XVII, No. 3, and also in the "Proceedings of the Society of Naval Architects and Marine Engineers," November, 1902. (Prize essay by Mr. Lovekin.)

The object of this article, in addition to giving the performance of the assistant cylinders, and comparing the performance of assistant and balance cylinders, installed on similar engines and working under the same conditions, is to describe a practical method of analyzing the indicator cards and working out the results. The calculations, diagrams and curves are given in full, so that they may be used as a form for future calculations and thus save much unnecessary work.

As the *Washington* is the first naval vessel to which Lovekin assistant cylinders have been fitted, as the power of the engines is much greater, and as the forces acting on the valve gear are greater than on any other engine to which these cylinders have been fitted, the results of the performance of these cylinders should be of unusual interest.

OBJECT OF THE ASSISTANT CYLINDER.

The object of the assistant cylinder is to *float* the valve gear, so as to relieve it of the load under which it works. The reasons for relieving this load are obvious. At the high revolutions the valve gear works under an enormous load, which is due principally to the inertia of the heavy moving parts. This load on the valve gear causes a great deal of friction, a great deal of wear on the eccentric bearings, and consequently frequent overhauling and repairing, which involves considerable expense.

Hence, the object of the assistant cylinder is to relieve the valve gear of the load under which it works, in order to reduce the wear on the bearings and the ensuing expenses. How well they fulfill this object can be seen from the curves of loads.

ACTION OF THE STEAM IN THE CYLINDER.

To clearly understand the action of the steam, refer to Figs. 1, 2 and analysis of indicator cards. Care should be

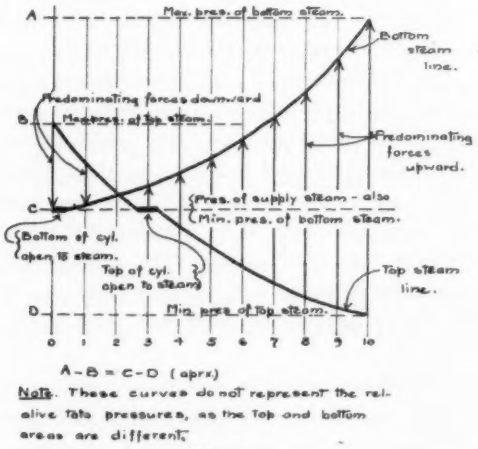

A
Max. pres. of bottom steam.
Bottom steam line.
Predominating forces downward
B
Max. pres. of top steam.
Predominating forces upward.
C
Pres. of supply steam - also
Min. pres. of bottom steam.
Bottom of cyl. open to steam.
Top of cyl. open to steam.
Top steam line.
D
Min. pres. of top steam.
0 1 2 3 4 5 6 7 8 9 10
A-B = C-D (aprx)
Note. These curves do not represent the relative total pressures, as the top and bottom areas are different.

Fig. 1.

taken not to confuse the top steam-pressure line of the indicator card with the line of "top steam plus gravity" of the diagrams.

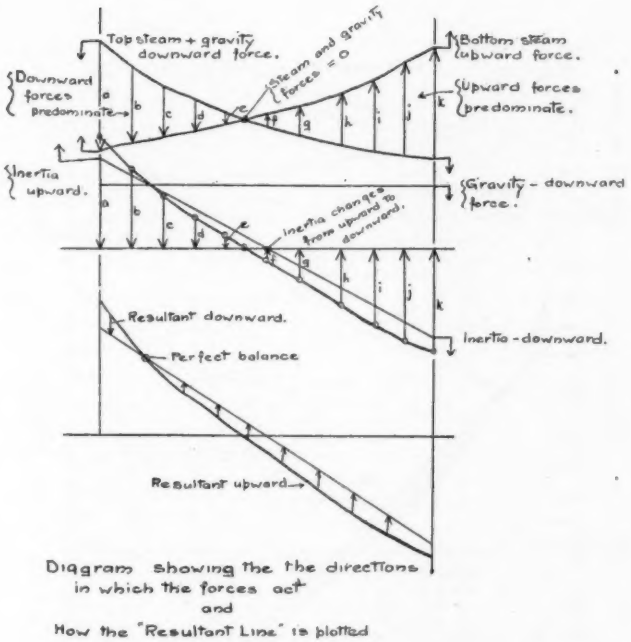
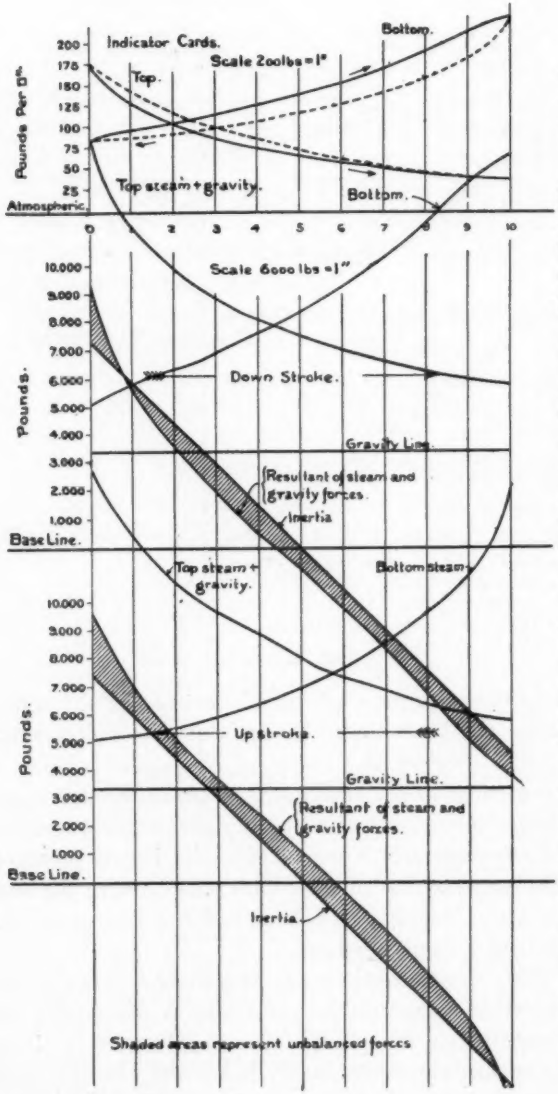

Top steam + gravity downward force.
Steam and gravity forces = 0
Bottom steam upward force.
Downward forces predominate.
Upward forces predominate.
Inertia upward.
Gravity - downward force.
Inertia changes from upward to downward.
Resultant downward.
Perfect balance
Inertia - downward.
Resultant upward.
Diagram showing the the directions in which the forces act and How the "Resultant Line" is plotted

Fig. 2.

To clearly illustrate the working of the cylinder the action of the steam will be followed throughout a complete revolution.

At the beginning of the stroke, the steam in the upper end of the cylinder is compressed to its highest pressure and the steam at the bottom end is expanded to its lowest pressure, which is the same as that of the supply steam. Also the supply port is open beneath the piston, and the bottom of the cylinder is open to supply steam.

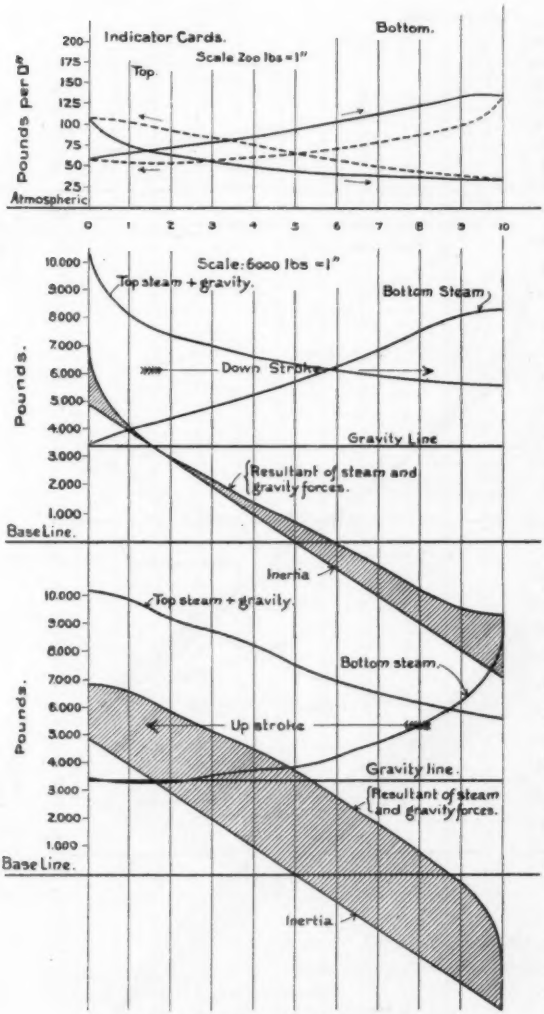
As the valve descends the steam under compression in the top of the cylinder expands and acts with a continually decreasing pressure on the piston. The supply port is closed by the piston and the steam supply below the piston is shut off. The steam in the bottom end of the cylinder is being



ANALYSIS OF INDICATOR CARDS, LOVEKIN IMPROVED ASSISTANT CYLINDER, U. S. S. "WASHINGTON."

Cylinder, Starboard H.P.
R. P. M., 124.
Links, full gear.

Per cent. of forces balanced—up-stroke, 79.5
down-stroke, 77.3
average, 78.4

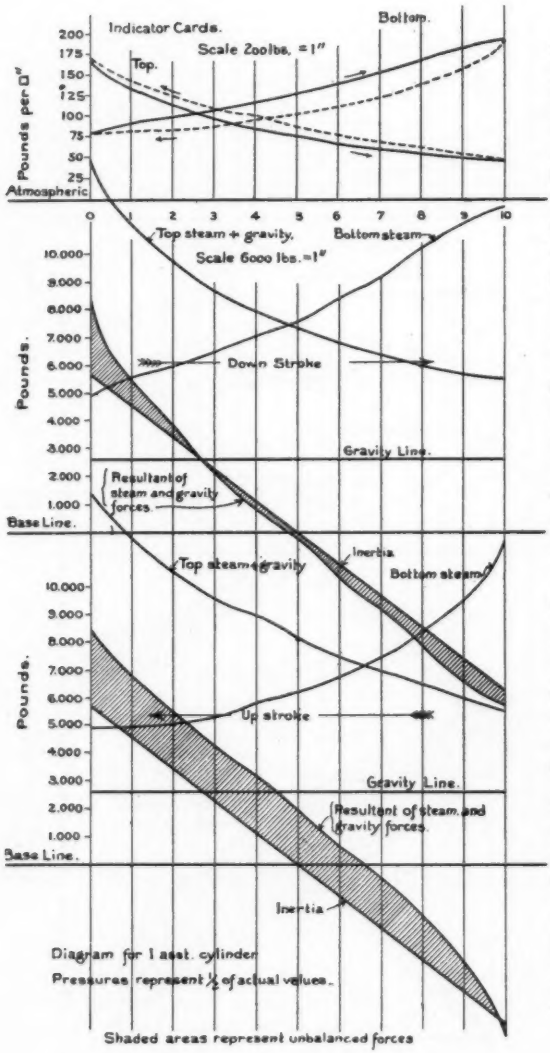


Shade areas represent unbalanced forces.

ANALYSIS OF INDICATOR CARDS, LOVEKIN IMPROVED ASSISTANT CYLINDER, U. S. S. "WASHINGTON."

Cylinder, Starboard H.P.
R.P.M., 101.
Links, full gear.

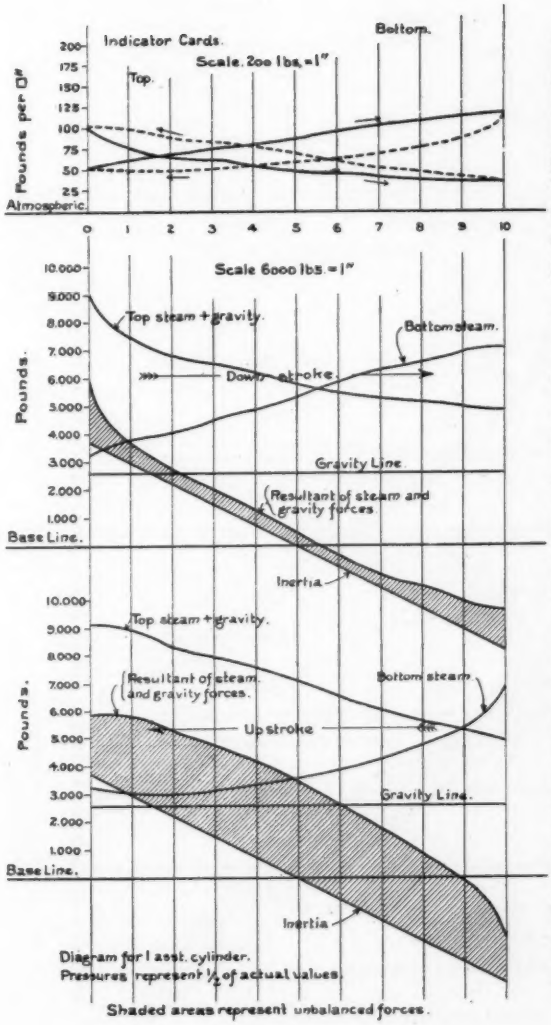
Per cent. of forces balanced—down-stroke, 75.0
up-stroke, 12.1
average, 43.6



ANALYSIS OF INDICATOR CARDS, LOVEKIN IMPROVED ASSISTANT CYLINDER, U. S. S. "WASHINGTON."

Cylinder, Starboard I.P.
R.P.M., 124.
Links, full gear.

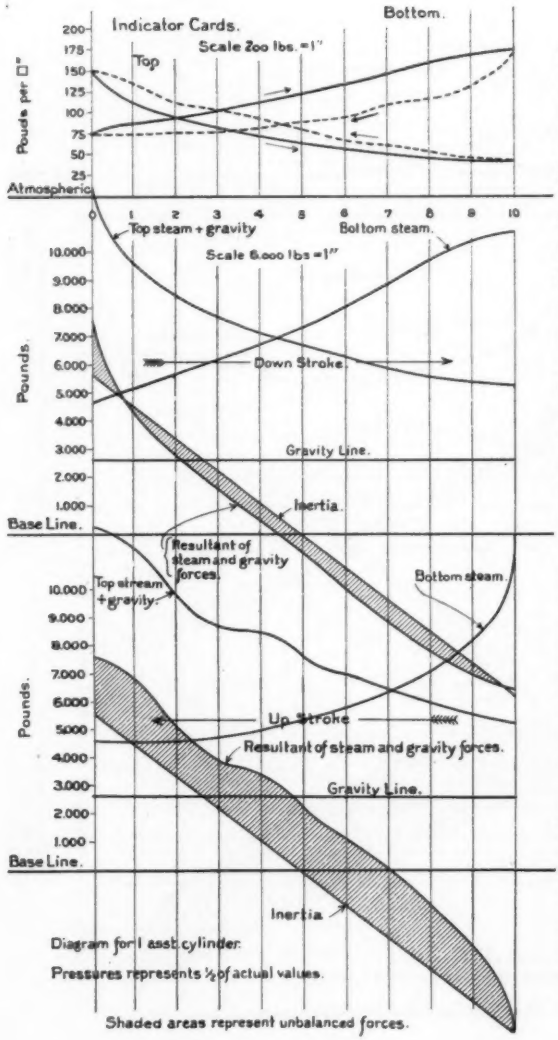
Per cent. of forces balanced—down-stroke, 80.0
up-stroke, 47.3
average, 63.6



ANALYSIS OF INDICATOR CARDS, LOVEKIN IMPROVED ASSISTANT CYLINDER, U. S. S. "WASHINGTON."

Per cent. of forces balanced—down-stroke, 70.7
up-stroke, 105.7
average, 32.5

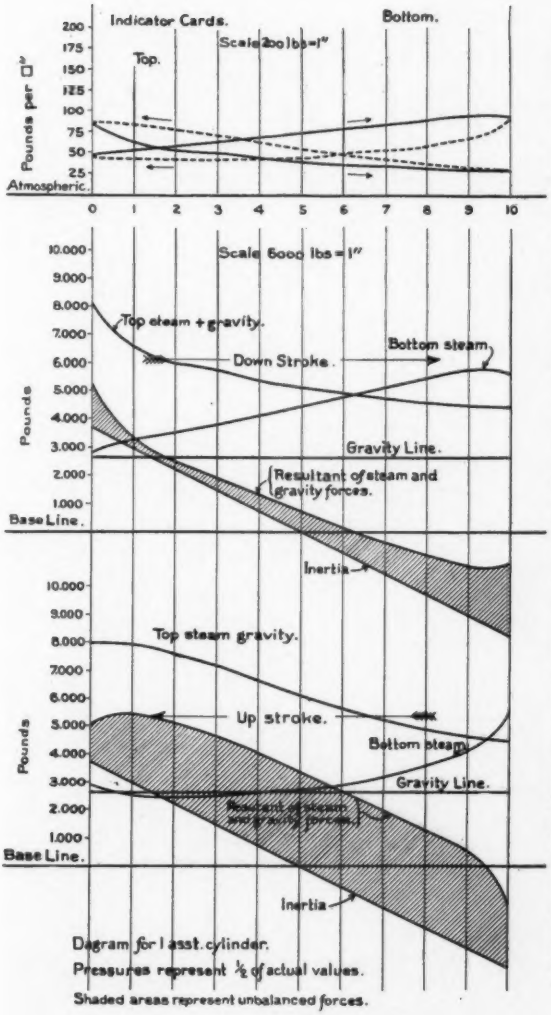
Cylinder, Starboard I.P.
R.P.M., 101.
Links, full gear.



ANALYSIS OF INDICATOR CARDS, LOVEKIN IMPROVED ASSISTANT CYLINDER, U. S. S. "WASHINGTON."

Per cent. of forces balanced—down-stroke, 81.0
up-stroke, 41.8
average, 61 4

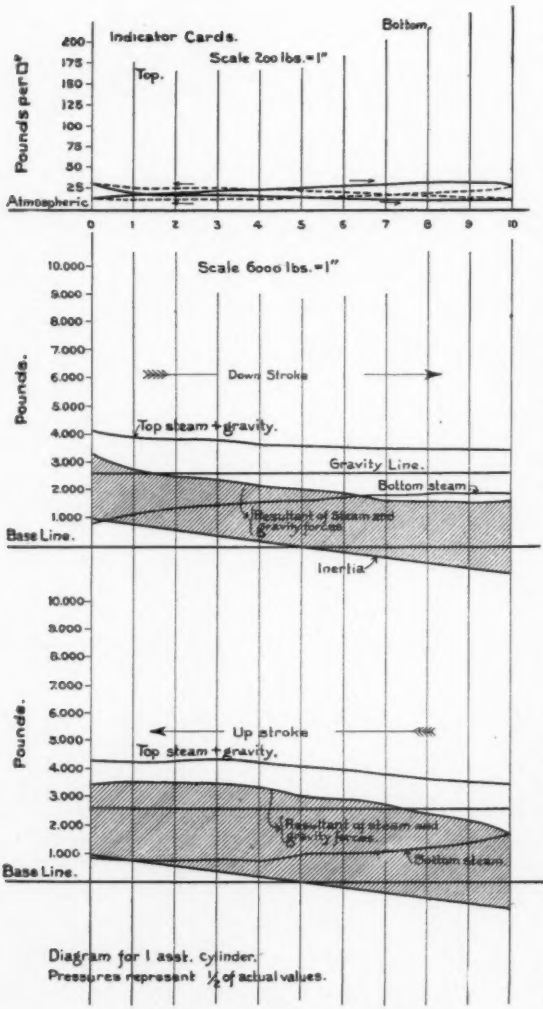
Cylinder, Starboard L.P.
R.P.M., 124.
Links, full gear.



ANALYSIS OF INDICATOR CARDS, LOVEKIN IMPROVED ASSISTANT CYLINDER, U. S. S. "WASHINGTON."

Cylinder, Starboard L.P.
R.P.M., 100.
Links, full gear.

Per cent. of forces balanced—down-stroke, 63
unbalanced—up-stroke, 109
balanced—average, 27



ANALYSIS OF INDICATOR CARDS, LOVEKIN IMPROVED ASSISTANT CYLINDER, U. S. S. "WASHINGTON."

Per cent. of forces balanced—down-stroke, 16.0
unbalanced—up-stroke, 118.5
unbalanced—average, 101.5

Cylinder, Starboard L.P.
R.P.M., 90.
Links, full gear.

compressed, and acts with a continually increasing pressure on the piston. The relations of the pressures acting on the piston can be readily understood by examining the indicator cards and Fig. 1.

The piston descends until it has traveled .27 stroke. At this point the steam in the upper end has expanded until it is equal to the supply. (This is theoretically the case, but actually the pressure is slightly below that of supply, due to the loss of heat through radiation.) The supply port then opens above the piston and the top of the cylinder is open to supply. The port is closed at .33 stroke, and the steam above the piston continues to expand until the minimum pressure is reached at the bottom of the stroke.

The bottom steam is compressed from the time the supply port shuts the steam off beneath the piston at .04 stroke until the maximum pressure is reached at the bottom of the stroke.

The piston now starts upward from the bottom of the stroke, the bottom steam expanding and the top steam being compressed. When the piston has traveled .67 stroke the top steam pressure has been compressed until it is equal to that of the supply. The supply port is opened above the piston at this point, and closed at .73 stroke. The top steam, which is being compressed, now rises above the pressure of the supply, reaching the maximum at the top of the stroke. The bottom steam expands until at .96 stroke its pressure is equal to that of supply. At this point the supply port opens beneath the piston. The minimum pressure of the steam below the piston is at this point, and is equal to the pressure of supply.

The steam consumption of the assistant cylinder is practically nothing, as the energy expended by the steam in expanding is returned when it is compressed. The very small steam consumption is due to radiation and the consequent condensation.

Note the fact that the bottom and top steam pressures at their maximum and minimum values differ by about the same

amount. (Not exactly the same, due to the fact that the expansion and compression curves differ slightly, but for the purpose of explanation they are assumed to be the same.) As these curves are approximately the same, the difference in the pressures above and below the piston at the same distances from the points of maximum pressures is always approximately constant and is equal to the difference of the maximum pressures. This constant difference is necessitated by the constant downward force of gravity and by the decreased area of the piston due to the rod.

A clearer way of putting it is this. If the gravity and decreased area of the bottom of the piston were compensated by a constant upward pressure in an ordinary balance cylinder, then only the inertia would be balanced in the assistant cylinder. As the inertia forces are the same at the top and bottom of the stroke, the compressions in the top and bottom of the cylinder would be the same, and there would be no difference in the relative action of the steam above and below the piston.

The compensation of the constant downward forces and the varying inertia forces is done in the same cylinder, in the case of the assistant cylinder; the constant downward forces being compensated by the constant relative difference between the top and bottom steam lines, and the inertia being compensated by compression as described above.

RELATIVE ACTION OF THE STEAM FORCES IN THE CYLINDER TO THE FORCES ACTING ON THE VALVE GEAR.

The action of the assistant cylinder may be compared to that of a spring dash-pot. The steam in the cylinder is the elastic medium, which, when being compressed, acts as a spring.

The steam in the cylinder so acts (governed by the laws of compression and expansion of steam) that the "predominating force" in the cylinder is always approximately equal to the forces acting on the valve gear, and in an opposite direc-

tion. (In referring to the steam under compression, for clearness, it is assumed that the steam under compression is the only force acting on the balance piston. Actually there is a steam pressure on each side of the piston, but it is only necessary to consider the *difference* in the two pressures, calling this the "predominating force" acting on the balance piston.) Thus the forces acting on the valve gear are neutralized, the valve gear is "floated," and the loads on it are relieved.

At the beginning of the downward stroke the inertia force is at its maximum, and inertia and gravity act in *opposite* directions. The "predominating force" at this point is approximately equal to the *difference* of these two forces and is in an opposite direction.

In this case, Predominating force=I—G. In the diagrams this is expressed thus: I=(top steam+gravity)—bottom steam. Thus: I—G=top steam—bottom steam=the predominating force.

As the valve descends, the forces acting on the valve gear are constantly decreasing; the predominating force is also decreasing at the same rate and is always approximately equal to these forces.

The inertia decreases until at .27 stroke it is equal to the gravity force, and, being in opposite directions, the forces are neutralized, and there is no force acting on the valve gear. At this point the predominating force is approximately nil.

By examining Fig. 1 and page 910, it will be seen that the top and bottom steam lines, down stroke, cross at .2 stroke, but this is accounted for by the fact that the area of the bottom of the piston is less than the top. Before this point was reached the forces acting on the valve gear were upward forces, and the predominating forces downward forces.

After passing this point the direction of the forces acting on the valve gear changes from upward to downward, and also the predominating forces change from downward to upward. (This can be more clearly understood from the

diagram, by noting the point at which the top and bottom steam lines of the indicator card cross, and the point at which the inertia line crosses the gravity line.) From this point, the forces acting on the valve gear continually increase until they reach the maximum at the bottom of the stroke. The predominating force increases at the same rate, and is always approximately of the same value until it reaches the maximum at the end of the stroke.

At the bottom of the stroke the inertia force again reaches its maximum, and it will be noticed that inertia and gravity are now acting in the same direction. At this point the forces acting on the valve gear reach their maximum and are in a downward direction.

CONSTRUCTION AND PIPING.

(See description, Vol. XVI, No. 3, and Fig. 3.)

Cylinder.—The assistant cylinder and the valve-chest cover are cast in one. The sides and top of the cylinder are lagged.

Steam-Supply Port.—An annular steam-supply port is located about the middle of the cylinder. This port opens into an annular space into which the steam-supply pipe is led.

Piston.—The piston is a little different from ordinary pistons, the cylindrical surface of the piston being extended above the conical part and connected to it by four webs. Just above the conical part of the piston an annular space is cut for the purpose of admitting steam above the piston. The piston is fitted with three cast-iron snap rings as shown.

Piston Rod.—The balance-piston rod is part of the valve stem, both being made from one piece of metal.

Packing.—Between the assistant cylinder and the valve chest the rod is packed with U. S. metallic packing.

Piping.—The steam-supply pipe leads into the annular steam port of the cylinder. All of the cylinders are supplied with steam at the same pressure from a differential valve on the steam-chest side of the throttle valve. (Thus steam is ad-

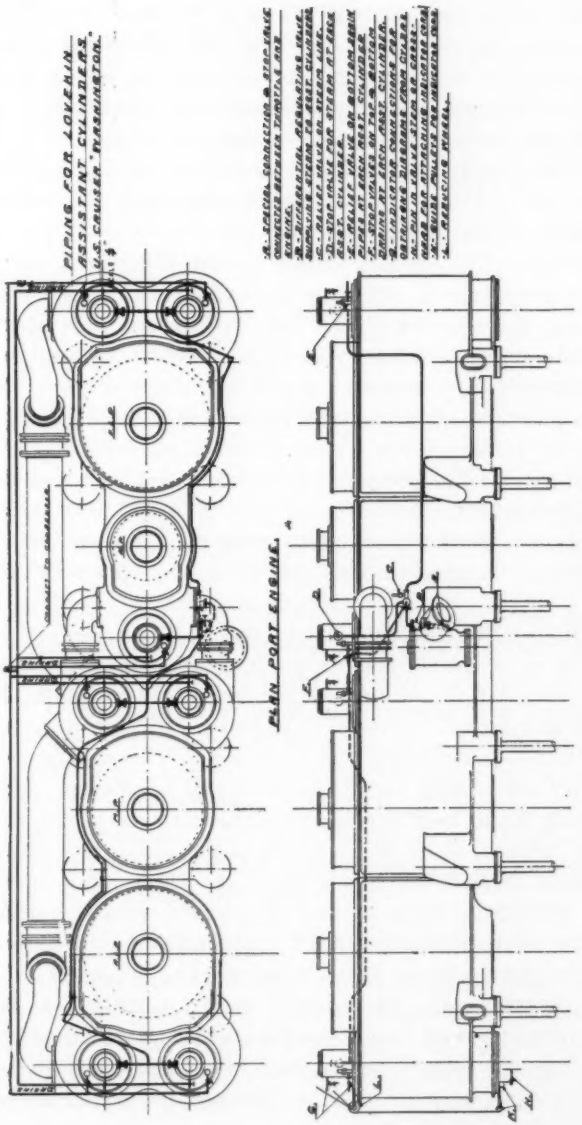

PIPING FOR LOVEKIN ASSISTANT CYLINDERS. U.S. CRUISER "WASHINGTON."
PLAN PORT ENGINE.
ELEVATION LOOKING OUTBOARD.

Fig. 3.

mitted to *all* of the assistant cylinders at once.) A relief valve is fitted on the supply pipe near the differential valve.

Drains.—The drain pipes are led into the top and bottom of the cylinder, and are fitted with valves. A relief valve is fitted on the drain pipe near the cylinder.

Both the supply and drain pipes have couplings at the cylinder, so that they can be disconnected when the valve-chest cover is taken off.

Differential Valve.—The object of the differential valve is to supply steam to the assistant cylinders at a pressure lower than the pressure in the H.P. valve chest from which the supply steam is taken; also to regulate the pressure so that a constant ratio between the pressure of the steam in the H.P. valve chest and the supply-steam pressure is always maintained. (The reason for the latter is as follows: The inertia varies as the square of the revolutions, and also the receiver pressure varies as the square of the revolutions. Consequently the inertia varies as the receiver pressure, and if the inertia is balanced at one speed it will be balanced at all speeds.) NOTE.—In designing these assistant cylinders it was assumed that the pressure in the H.P. receiver varies as the square of the revolutions, but it does not exactly, owing to the fact that the steam has not begun to do work. In the first and second receivers (I.P. and L.P.), however, this relationship is maintained very accurately. This was proved by laying off several curves under actual conditions. Had the supply steam been regulated by the pressure in the first receiver, instead of by the pressure in the H.P. valve chest, the balance at the lower speeds would have been much better.

The differential valve consists of a piston with the steam from the valve chest acting on a small area and the supply steam acting on a larger area. When the two pressures differ by the constant ratio the piston is exactly balanced. If the supply-steam pressure is too low the piston moves and admits steam into the supply piping, and when the supply-steam pressure rises above its relative pressure the piston moves back and shuts off the steam.

The ratio of pressures depends on the areas subjected to valve-chest pressure and supply-steam pressure.

Let

p_v = pressure per square inch in valve chest.

p_s = pressure per square inch of the supply steam.

a_v = area of the piston subjected to the pressure of the steam in the valve chest.

a_s = the area of the piston subjected to the pressure of supply steam.

Then

$$a_v : a_s = p_s : p_v.$$

Indicator Gear.—The piping and valves for the indicators are the same as for ordinary cylinders. The motion of the valve is transmitted to the drum from a pin on the cross-head to which indicator cord is attached. From this pin the cord leads to a guide pulley on the bottom of the valve chest vertically above the pin, then to a fair lead pulley and up to the top of the valve chest to a reducing wheel. From the reducing wheel the drum chord is led.

INSTRUCTIONS FOR OPERATING THE LOVEKIN IMPROVED ASSISTANT CYLINDER FOR VALVE GEARS.

To Warm Up.—(1) Open top and bottom drain valves. (2) Open steam admission valve a sufficient amount for warming. (3) Leave all valves in these positions until engine has begun to turn over; then close drain valves and open admission valve full open. (4) Keep all valves in these positions while engine is running. NOTE.—It is advisable to open the drain valves once every watch to make certain that there is no accumulation of water.

TAKING INDICATOR CARDS.

The same rules that apply to taking cards from ordinary cylinders apply to taking cards from assistant cylinders.

Before taking cards the down-stroke line and the up-stroke line for both the top and the bottom ends of the cylinder should be determined and marked on cards for reference.

(Under normal conditions the compression line is above the expansion line, but this may not be so if the cylinder is not functioning properly, and in the latter case it would cause serious errors in determining the balance.)

The following data should be entered on the card: Time of taking, R.P.M. at this time, scale of the indicator spring, the assistant cylinder from which the card was taken, and the date.

COMPUTATIONS AND DIAGRAMS.

In working up the cards from the assistant cylinders blue prints* were made of the forms for the diagrams and computation sheet. Where a number of cards are to be worked up it is easier to do it in this way.

The order of work is as follows: (1) The data are worked up and tabulated in convenient form for reference, as in the "Data Summary." (2) The diagrams are constructed and the data worked up on the computation sheet. (3) The following curves are constructed from the data on the computation sheet: (*a*) "Maximum Loads on the Valve Gear;" (*b*) "Mean Loads on the Valve Gear."

Laying Out the Diagrams.—First, fill in the data from the "Data Summary" and other blanks, such as the cylinder and R.P.M., on both the diagram and computation sheets.

Indicator Cards.—Draw ordinates from the atmospheric line tangent to each end of the "card" (indicator diagram). Divide the atmospheric line into 10 parts between these ordinates, and draw in the ordinates. Number these ordinates 0, 1, 2, . . . 10. Mark the compression and expansion lines and the direction the piston is moving as shown. NOTE.—Under normal conditions, the upper line is the compression line and the lower line the expansion line.

Begin with the *top* end of the cylinder and *down* stroke; measure the pressures at the ordinates and fill in columns 1, 3, 5 and 7 of the table in the order they come. NOTE.—Care should be taken not to confuse the *up* with the *down* stroke,

* The Van Dyke process was used in order to obtain dark lines and light ground.

nor the *expansion* line with the *compression* line. *Compression* at one end always goes with *expansion* at the other.

Multiply the pressures by the constant, filling in columns, 2, 4, 6 and 8, Compute the inertia as shown. Lay out the diagram as follows:

(1) Draw in the gravity line } for both down
(2) Draw in the inertia line } and up strokes.

For the *down* stroke—

(3) Lay off the inches to scale of the *top* steam *up* from the *gravity* line on the corresponding ordinates, marking the points.

(4) Draw in the line of *top steam+gravity.*

(5) Lay off the inches to scale of the *bottom* steam *up* from the *base* line on the corresponding ordinates, marking the points.

(6) Draw in the line of *bottom steam.*

Do the same for the *up* stroke.

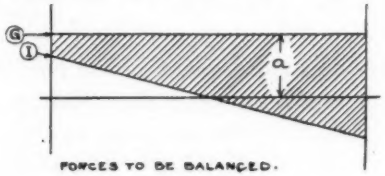
Combining the Lines and Obtaining the "Resultant."—(See the Diagram of Forces, Fig. 2.)—NOTE. When the line of *top steam+gravity* is *above* the *bottom steam* line the resultant force is downward, and the ordinates are laid off *upward* from the *base* line. When the *bottom* steam line is *above* the resultant force is upward, and the ordinates are laid off *downward* from the *base* line.

Measure the distances on the ordinates intercepted between the *bottom steam* line and the line of *top steam+gravity.* Lay these distances off from the *base* line on the same ordinates (up or down as explained above) and mark the points. Draw in the resultant line.

Per Cent. of Forces Balanced.—Measure the portions of the ordinates intercepted between the inertia line and the resultant line, *regardless* whether they are *up or down* from the inertia line, and fill in columns B and C. Calculate B and C—mean ordinates of shaded area.

In calculating the mean ordinate of the area representing the forces to be balanced, two cases present themselves.

See Fig. 4. When the point I of the inertia line corresponds with or is below point G of the gravity line, the mean ordinate of the forces to be balanced is the same as *a*, the ordinate of the gravity. (The unshaded triangle above the base line is equal to the shaded triangle below the base line; hence, the shaded area is the same as the area of the rectangle included



Case 1.

Fig. 4.

between the base and gravity lines and the end ordinates. The mean ordinate of this area is equal to the ordinate of the gravity force.

See Fig. 5. When point I is above point G the mean ordinate of the shaded area is found as follows:

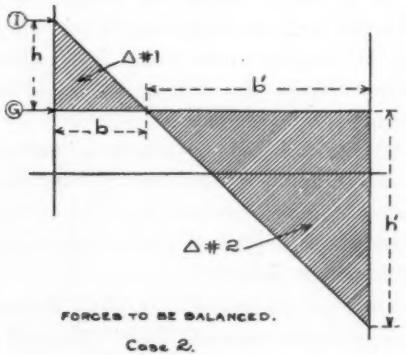


Fig. 5.

Find the areas of the two triangles △ No. 1 and △ No. 2 (½b×h). Add the areas of the two triangles and divide by the sum of the bases, which gives the mean ordinate. As the

mean ordinate of the forces to be balanced is the same for both the down and the up strokes, it is necessary to calculate it only for the down stroke.

Let

A = mean ordinate of forces to be balanced.
B = mean ordinate of shaded area, down stroke.
C = mean ordinate of shaded area, up stroke.

Then the per cent. of forces balanced, down stroke $= \frac{A-B}{A}$.

Then the per cent. of forces balanced, up stroke $= \frac{A-C}{A}$.

Loads on the Valve Gear.—The maximum load to be balanced is measured from the diagram at the bottom of the stroke, and is the sum of gravity and maximum inertia. The maximum unbalanced load on the valve gear is the maximum ordinate of the shaded area times the scale.

The mean load to be balanced is A (the mean ordinate of forces to be balanced) times the scale. The mean load unbalanced is the mean of B and C (the mean ordinate of the shaded area for a complete revolution) times the scale.

The curves are constructed as shown in Figs. 6 and 7, from the data on the computation sheets under the heading "Loads on Valve Gear."

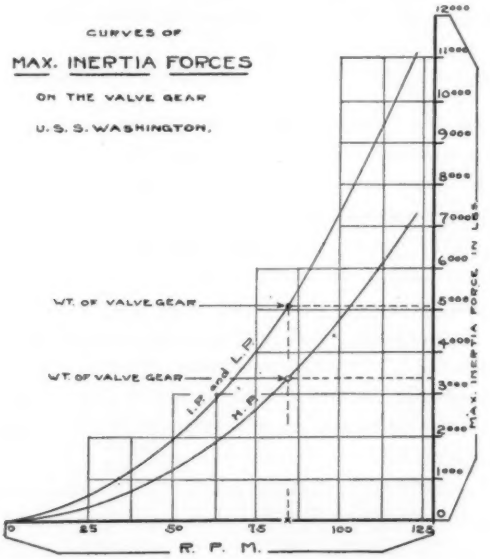
After laying out the scales for these curves the lines representing the weight of the valve gear are drawn. At 0 revolutions the load on the valve gear is due to the weight of the valve gear alone; hence the curves start at these points and are drawn from these points through the points platted from the computation sheet.

In the curves of "Mean Loads" it will be noticed that the curves of "Loads on valve gear to be balanced" is a straight line from 0 to 84 revolutions. This is due to the fact that the maximum inertia is less than the gravity up to this point.

The "Lower speed limit" marked on the curves is the limit of speed below which the engines are not likely to be run in cruising.

PERFORMANCE OF THE ASSISTANT CYLINDERS.

Without considering the results of the analysis of the indicator cards, the performance of the assistant cylinders, judging from the easy working of the valve gear, was excellent. There was no noise or jar in the working of the valve gear, and especially at the bottom of the stroke, where the forces acting on the valve gear are considerable.



The performance of the assistant cylinders for various speeds is shown in the curves of loads on the valve gear, Figs. 6 and 7. As these curves clearly show the excellence of the performance, very little need be said on this subject.

BALANCE AT FULL SPEED.

For the H.P. cylinder the "mean load" is less than $\frac{1}{3}$ of the load on the valve gear when the engine is at rest. By the use of the assistant cylinder the "mean load" is reduced from 4,600 pounds to 958 pounds. The "maximum load" is only

⅔ of the load on the valve gear when the engine is at rest. By the use of the assistant cylinder the "maximum load" is reduced from 10,600 pounds to 2,200 pounds.

For the I.P. and L.P. cylinders the "mean load" is about ½ of the load on the valve gear when the engine is at rest.

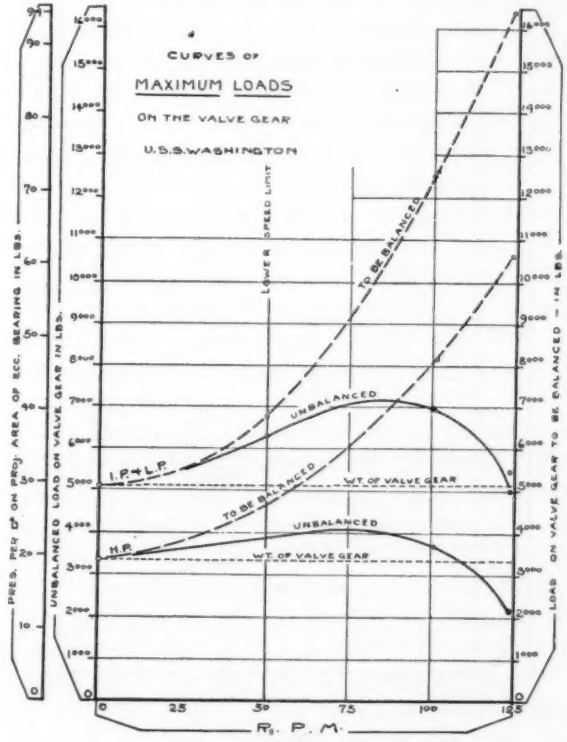


Fig. 6.

By the use of the assistant cylinder the "mean load" is reduced from 6,800 pounds to 2,500 pounds. The "maximum load" is about the same as the load on the valve gear when the engine is at rest. By the use of the assistant cylinder the "maximum load" is reduced from 16,400 pounds to 5,100 pounds.

As the valve gear is subjected to the greatest loads at this speed, the balance should be the best, and, by examining the curves, this is found to be the case.

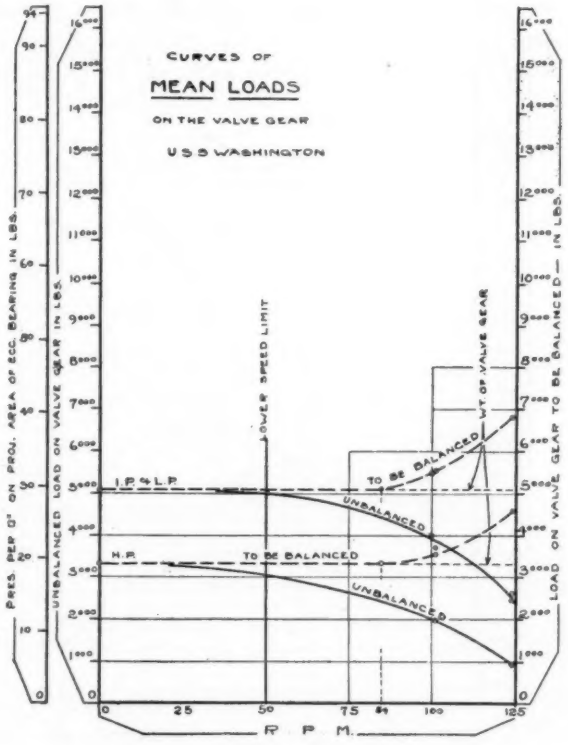


Fig. 7.

BALANCE AT CRUISING SPEEDS.

The revolutions for cruising speed will be between 65 and 80. The balance for cruising speeds, although not so good as at full speed, is very satisfactory.

For the H.P. the "mean loads" are from $\frac{4}{5}$ to $\frac{3}{4}$ of the load on the valve gear when the engine is at rest, and the

"maximum loads" are reduced from 5,300—6,500 pounds to about 4,100 pounds.

For the I.P. and L.P. the "mean loads" are .98 to .9 of the load on the valve gear when the engine is at rest, and the "maximum loads" are reduced from 7,900—10,000 pounds to 6,800—7,200 pounds. At these speeds it will be noticed that the "maximum unbalanced loads" are the greatest. As most of the cruising of this vessel will be done at these speeds, the balance should have been better, and would have been better had the supply-steam pressure been controlled by the steam pressure in the receiver instead of by the steam in the H.P. valve chest. In other vessels, where the supply steam is controlled by the receiver pressure, the performance of the assistant cylinders has been much better.

It will be noticed that with balance cylinders the load on the valve gear is evenly distributed throughout the complete revolution. This is an important consideration, as it obviates the uneven load and excessive pressure on the bearing at the bottom of the stroke.

Theoretically the balance at all speeds obtained by using the assistant cylinder is nearly perfect. The one difficulty is regulating the pressure of the supply steam. Even when the pressure of the supply steam is regulated by the pressure in the first receiver the regulation is not perfect. Various conditions, such as changing the setting of the independent cut-off, alter the pressure in the receiver, so that this pressure does not always bear a fixed relation to the revolutions.

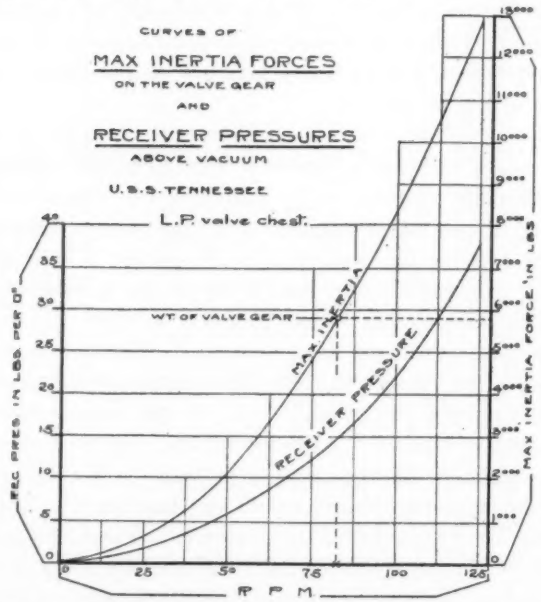
There is no reason why the balance should not be almost perfect with the proper regulation of the steam supplied to the assistant cylinders. Considering the fact that the automatic regulation by the receiver pressure is not as close as it should be, better results would be obtained if a hand adjustment were provided so as to correct the pressure as regulated by the receiver. This could be done by using a differential valve, the same as used on the *Washington,* but with a spring connected with the valve in such a way that the spring

would add to or decrease the pressure on the controlling end of the valve and increase or decrease the supply-steam pressure as desired.

From the cards analyzed a table could be made showing the supply-steam pressure and the revolutions corresponding. By adjusting the supply-steam pressure to correspond with the revolutions an almost perfect balance could be obtained over a large range of speed.

A COMPARISON OF THE PERFORMANCE OF THE LOVEKIN ASSISTANT CYLINDER ON THE U. S. S. *WASHINGTON* WITH THE ORDINARY BALANCE CYLINDER ON THE U. S. S. *TENNESSEE*.

It will be seen that the weight of the valve gear on the *Washington* is 700 pounds lighter than on the *Tennessee*. This difference should not be considered, as the reduction in



the weight on the *Washington* was due to the use of the assistant cylinder and the resulting small values of the maxi-

mum loads on the valve gear. By examining the curves of maximum loads it will be seen that in the valve gear on the *Tennessee,* although heavier, the metal is under a greater strain than on the *Washington.*

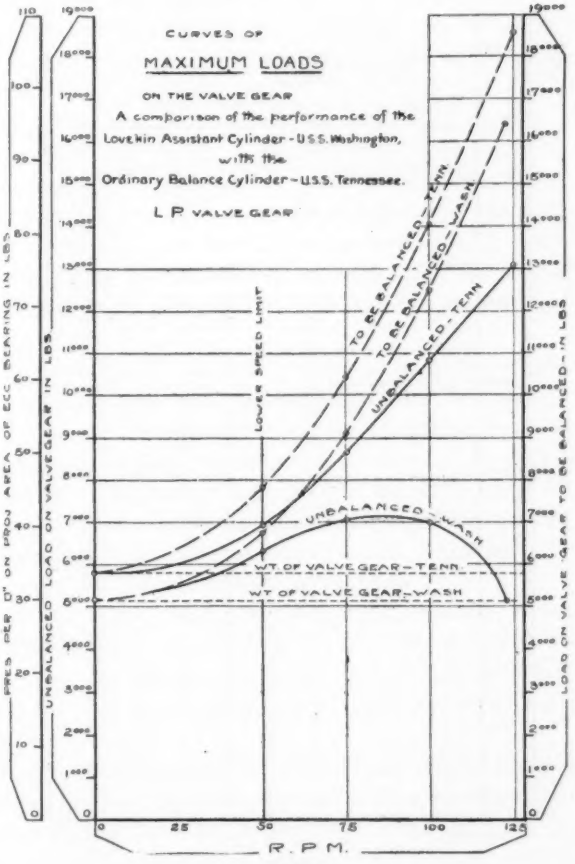

CURVES OF
MAXIMUM LOADS
ON THE VALVE GEAR
A comparison of the performance of the
Lovekin Assistant Cylinder - U.S.S. Washington,
with the
Ordinary Balance Cylinder - U.S.S. Tennessee.
L P VALVE GEAR
LOWER SPEED LIMIT
TO BE BALANCED - TENN
TO BE BALANCED - WASH
UNBALANCED - TENN
UNBALANCED - WASH
WT OF VALVE GEAR - TENN
WT OF VALVE GEAR - WASH
LBS PER SQ IN ON PROJ AREA OF ECC BEARING IN LBS
UNBALANCED LOAD ON VALVE GEAR IN LBS
TO BE BALANCED LOAD ON VALVE GEAR IN LBS
R. P. M.

Fig. 8.

For comparison the curves of maximum and mean loads for both the *Washington* and *Tennessee* have been plotted on the same diagrams. See Figs. 8 and 9.

The comparison is made using the assistant and balance cylinders on the L.P. valve gear. The engines of the *Washington* and *Tennessee* are practically similar, and the horsepower and revolutions approximately the same. The data

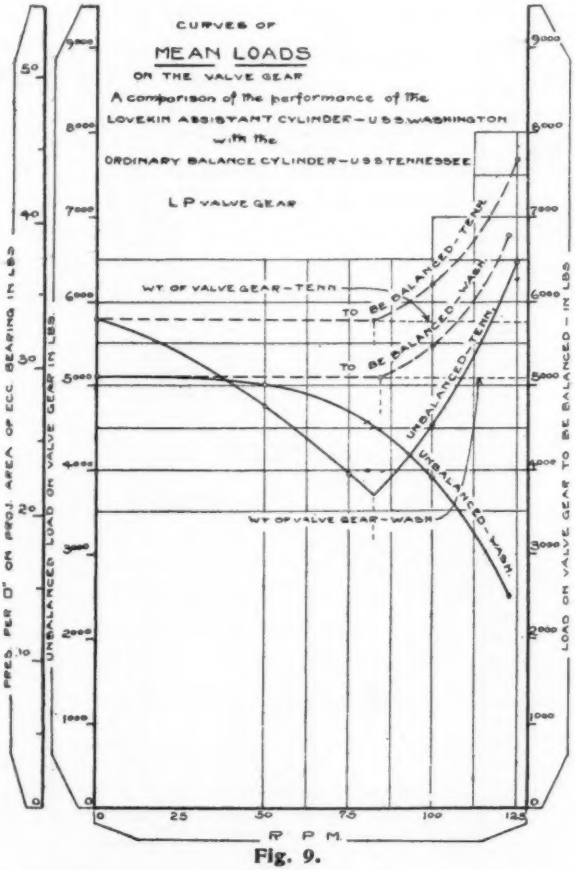

CURVES OF
MEAN LOADS
ON THE VALVE GEAR
A comparison of the performance of the
LOVEKIN ASSISTANT CYLINDER – U.S.S. WASHINGTON
with the
ORDINARY BALANCE CYLINDER – U.S.S. TENNESSEE
L.P. VALVE GEAR
WT OF VALVE GEAR – TENN
TO BE BALANCED – TENN
TO BE BALANCED – WASH
UNBALANCED – TENN
UNBALANCED – WASH
WT OF VALVE GEAR – WASH
PRES. PER □" OR PROJ. AREA OF ECC. BEARING IN LBS
UNBALANCED LOAD ON VALVE GEAR IN LBS.
LOAD ON VALVE GEAR TO BE BALANCED – IN LBS
R P M.

Fig. 9.

used in these comparisons was taken during the official acceptance trials of the ships, so that the conditions for both the assistant and balance cylinders were the same.

The curves clearly show the performance of the two cylinders.

At full speeds—at which speeds accidents and breakdowns are most likely to occur, and consequently the speed at which the best balance is desired—the maximum unbalanced load on the *Washington's* valve gear was much less than that on the *Tennessee,* the former load being 5,000 pounds and the latter 13,080 pounds.

The mean load at full speed is only 2,500 pounds on the *Washington,* and on the *Tennessee* it is 6,480 pounds.

The curve of maximum loads on the *Washington* is much better than on the *Tennessee.* It is important to balance this maximum load at the bottom of the stroke, as the strain on the valve gear at this point is excessive. In this respect the assistant cylinder is far better than the ordinary balance cylinder.

The mean loads on the valve gear of the *Tennessee* decrease from the maximum until the minimum value is reached where I=G. As can be seen from the curve, the balance cylinder would work very well were it not for inertia. As soon as the inertia becomes greater than gravity the unbalanced load increases very rapidly, until at full speed the unbalanced load nearly equals the load to be balanced, and at full speed the best balance is desired and the worst obtained.

The curve for the ordinary balance cylinder crosses the assistant cylinder curve at 93 revolutions and continues below it for all practical cruising speeds. The difference between the two is not great, the greatest difference being 750 pounds.

It may be repeated here, that had the supply steam on the *Washington* been properly regulated these results would have been far better, and the superiority of the assistant cylinder over the ordinary balance cylinder shown more conclusively. Better results than these have been obtained, and there is every reason to believe that the best results can be greatly improved in speed regulation (regulation of supply steam

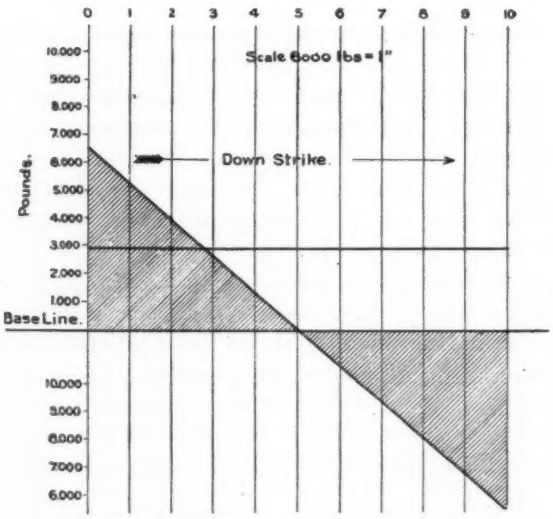


DIAGRAM SHEET, BALANCE CYLINDER, U. S. S. "TENNESSEE."

Cylinder, L.P. R.P.M., 125.

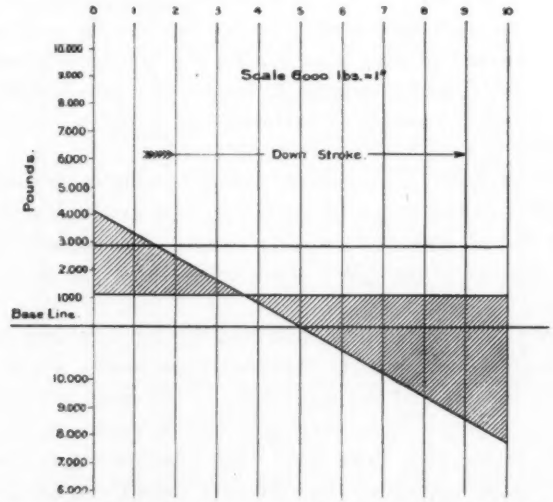


DIAGRAM SHEET, BALANCE CYLINDER, U. S. S. "TENNESSEE."

Cylinder, L.P. R.P.M., 100.

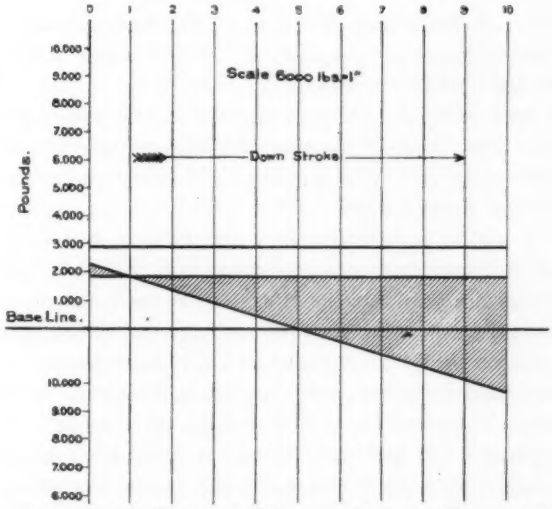


DIAGRAM SHEET, BALANCE CYLINDER, U. S. S. "TENNESSEE."

Cylinder, L.P. R.P.M., 75.

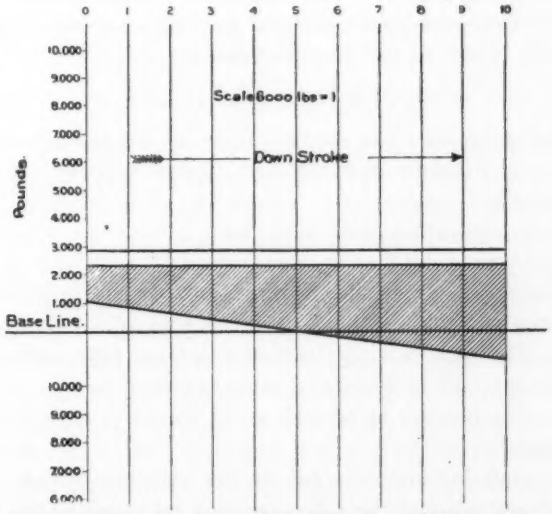


DIAGRAM SHEET, BALANCE CYLINDER, U. S. S. "TENNESSEE."

Cylinder, L.P. R.P.M., 50.

pressure). Even on the *Washington* the performance of the assistant cylinder on the H.P. valve gear was better than for the L.P. assistant cylinder.

The superiority of the performance of the assistant cylinder over the ordinary balance cylinder on similar engines working under the same conditions is demonstrated beyond question by these curves.

In the above comparison only the balance has been considered. In regard to steam consumption, with the assistant cylinder as designed for the *Washington* the only steam used is the small amount liquefied plus whatever may leak through joints, there being no connection with the exhaust. In the ordinary balance cylinder the loss from leakage is uncertain, as, unless the piston is perfectly tight, the steam will continually leak over, and such leakage is difficult to detect.

In regard to weight and space the assistant cylinder still has a large advantage due to using steam of higher pressure.

The conclusion that may be drawn is that the assistant cylinder is superior in every way to the balance cylinder, and where a balance cylinder could be used to advantage it would be much better to use the assistant cylinder.

ECONOMICAL FEATURES.

Disadvantages.—The disadvantages of assistant cylinders, which are the same as for balance cylinders are—

(1) Initial cost.

(2) Complication and more parts.

(3) More weight and space required.

Advantages.—The above disadvantages are more than compensated by the following advantages:

(1) The reduction of wear on the valve gear, and consequently the cost of maintenance and up-keep.

(2) The liability to breakdown in the valve gear is greatly reduced.

In comparing the economy of the assistant cylinder with the balance cylinder the following may be added to the above advantages:

(3) Smaller and lighter cylinders may be used.

(4) The initial cost is less in case of large cylinders.

The saving in maintenance is due to the reduced load on the valve gear, less friction and consequently less oil required and less wear on the bearing. Less oil means a direct saving in the cost of oil; less wear on the bearing means less overhauling and repairing and a smaller repair expense for the valve gear.

This is not only the theoretical economy, but it is also the actual, as has been borne out in practice. For instance, the S.S. *Texan,* on which these cylinders are fitted, has steamed over 150,000 miles, and the wear on the eccentric bearings has been practically nothing. It has not even been necessary to put in or take out a liner. The cost of overhauling and repairing the valve gear has been practically nothing. The other ships on which these cylinders are fitted have given results that are nearly as good as the above.

From the above, considering the extra cost of assistant cylinders, it appears that the use of assistant cylinders is amply justified by the fact that the saving in repair expenses soon exceeds the cost of installing the cylinders.

APPLICATION.

Merchant Vessels.—Assistant cylinders have been fitted on several merchant vessels—notably, the Pacific Mail steamships *Manchuria* and *Mongolia.* The primary consideration, whether to or not to fit assistant cylinders on merchant vessels is that of economy. From the results of the ships on which these cylinders are installed, and the performance of the assistant cylinders on the *Washington,* it seems that there is considerable economy in their use. In comparison with balance cylinders, it seems that for about the same or less cost assistant cylinders give better results and better economy. The conclusion, then, would be that it would be advantageous, from the standpoint of economy, to install assistant cylinders on merchant vessels.

Naval Vessels.—In naval vessels weight and space are of

primary importance. The feasibility of applying assistant cylinders to naval vessels has been demonstrated on the *Washington,* the *Kansas* and the *New Hampshire,* on which these cylinders are being fitted.

Taking into consideration the facts that very little extra weight is necessary and the reduced loads on the valve gear, there is every reason to believe that assistant cylinders are desirable for naval vessels.

In comparison with balance cylinders, assistant cylinders are much more desirable, as they are lighter and relieve the loads on the valve gear more efficiently. The latter is important, as it reduces the liability to breakdown from the valve gear and also reduces the overhauling necessary.

Stationary Engines.—Assistant cylinders are also applicable to stationary engines where the loads on the valve gear are considerable, and in many cases their use would be advantageous.

CONCLUSION.

Taking into consideration—

(1) The superiority of assistant cylinders over balance cylinders;

(2) The excellent performance of assistant cylinders as shown by test and by their performance in actual service and the ensuing economy from their use;

(3) That the initial cost of installing assistant cylinders is about the same as for balance cylinders;

(4) Their adaptability;

The conclusion is—

(1) Wherever the use of balance cylinders is desirable assistant cylinders would be better;

(2) That there is a reduction in expenses due to the use of assistant cylinders;

(3) That it is desirable for economical reasons to install assistant cylinders on merchant vessels;

(4) That it seems to be desirable, for the constant efficiency of the valve gear under hard service, to install assistant cylinders on naval vessels;

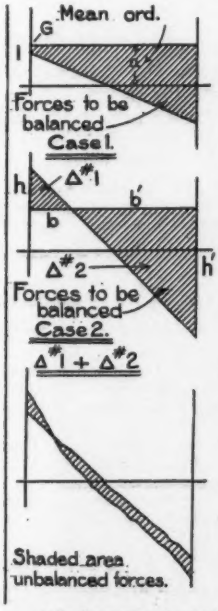
ANALYSIS OF INDICATOR CARDS.

Lovekin Improved Assistant Cylinder. U. S. S. *Washington*, April 12, 1906.

Cylinder, starboard H.P.; R.P.M., 124.

Computed by W. W. Smith.

Ordinates.		*Down stroke.* Pressure per sq. in.	*Down stroke.* Inches to scale.	*Up stroke.* Pressure per sq. in.	*Up stroke.* Inches to scale.	Ordinates.		*Down stroke.* Pressure per sq. in.	*Down stroke.* Inches to scale.	*Up stroke.* Pressure per sq. in.	*Up stroke.* Inches to scale.	*Ordinates of shaded area.* Down stroke.	Up stroke
		1	2	3	4			5	6	7	8	B	C
		Top.						Bottom.					
		Expansion.		*Compression.*				*Compression.*		*Expansion.*			
0	Const. $= \frac{A}{S} = .0212$	175	3.71	...	...	0	Const. $= \frac{A}{S} = .0203$	83	1.68	...	...	.73	.73
1		125	2.65	140	2.96	1		94	1.91	86	1.75	.09	.38
2		101	2.14	114	2.42	2		103	2.09	90	1.83	.30	.21
3		85	1.80	97	2.05	3		112	2.27	96	1.95	.33	.21
4		73	1.55	84	1.78	4		124	2.52	104	2.11	.33	.29
5		64	1.36	71	1.51	5		135	2.74	113	2.29	.25	.31
6		56	1.19	61	1.29	6		150	3.05	124	2.52	.25	.35
7		50	1.06	54	1.14	7		167	3.39	139	2.82	.25	.40
8		44	.93	46	.98	8		187	3.80	156	3.17	.30	.37
9		40	.85	40	.85	9		210	4.26	178	3.61	.35	.30
10		37	.78	...	...	10		226	4.59	...	...	.28	.28
											Sum =	3.46	3.83



FORCES ACTING ON VALVE GEAR.

$K = \log 9.67403$

$N^2 = 124^2 \quad \log 4.18684$

$\log 3.86087$

G = Gravity force 3,330 pounds.

I = Inertia force 7,259 pounds.

PER CENT. OF FORCES BALANCED.

Forces to be balanced *Case I.—When point I, is below point G, the mean ordinate of the forces to be balanced = a = ordinate of gravity line.

Mean ordinates.

A = mean ord. of forces to be balanced = 1.535 in.

B = mean ord. of shaded area down stroke = .314 in.

C = mean ord. of shaded area up stroke = .348 in.

Per cent. of forces balanced.

Forces balanced down stroke $= \frac{A - B}{A} = 79.5$ per ct.

Forces balanced up stroke $= \frac{A - C}{A} = 77.3$ per ct.

Forces balanced mean — for 1 revolution = 78.4 per ct.

LOADS ON VALVE GEAR.

To be balanced.	*Maximum loads.*	Pounds.
	Max. load on ecc. bearing (from diagram) =	10,620
	Max. pres. per □″ of proj. area of ecc. bearing =	61
Unbalanced.		
	Max. load on ecc. bearing (from diagram) =	2,190
	Max. pres. per □″ of proj. area of ecc. bearing =	13
To be balanced.	*Mean loads (during a complete revolution).*	
	Mean load on ecc. bearing (A × 3,000) =	4,605
	Mean pres. per □″ of proj. area of ecc. bearing =	26
Unbalanced.		
	Mean load on ecc. bearing $\frac{(B + C)}{2} \times 3{,}000$ =	958
	Mean pres. per □″ of proj. area of ecc. bearing =	6

* As the mean ordinates for both the Up and Down strokes are the same, it is necessary to calculate it only for the down stroke.

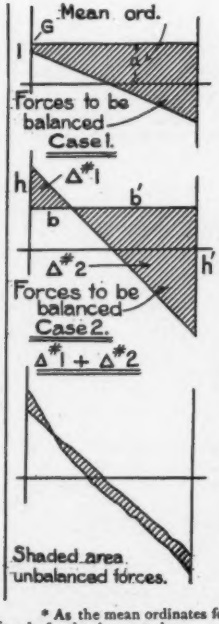
ANALYSIS OF INDICATOR CARDS.

Lovekin Improved Assistant Cylinder. U. S. S. *Washington*, April 12, 1906.

Cylinder, starboard L.P.; R.P.M., 50.

Computed by C. A. Gardiner. Checked by W. W. Smith.

Ordinates.		*Down stroke.*		*Up stroke.*		Ordinates.		*Down stroke.*		*Up stroke.*		*Ordinates of shaded area.*	
		Pressure per sq. in.	Inches to scale.	Pressure per sq. in.	Inches to scale.			Pressure per sq. in.	Inches to scale.	Pressure per sq. in.	Inches to scale.	Down stroke.	Up stroke.
		1	2	3	4			5	6	7	8	B	C
		Top.						Bottom.					
		Expansion.		*Compression.*				*Compression.*		*Expansion.*			
0	Const. $= \frac{A}{S} = .0212$	27	.57	...	...	0	Const. $= \frac{A}{S} = .0203$	14	.28	...	....	.78	.78
1		20	.42	26	.55	1		18	.36	13	.26	.66	.92
2		18	.41	26	.55	2		21	.42	13	.26	.64	.98
3		18	.41	26	.55	3		23	.47	14	.28	.65	1.04
4		17	.33	25	.53	4		25	.51	14	.28	.64	1.06
5		15	.32	23	.49	5		26	.53	16	.32	.66	1.02
6		14	.30	21	.44	6		28	.57	17	.34	.66	1.02
7		13	.28	19	.41	7		30	.61	18	.36	.66	1.04
8		13	.28	17	.33	8		32	.62	20	.41	.69	1 00
9		13	.28	15	.32	9		32	.62	22	.45	.74	.97
10		13	.28	...	...	10		30	.61	...	....	.80	.87
											Sum =	7.58	10.70



FORCES ACTING ON VALVE GEAR.

$K = \log 9.56321$

$N^2 = 50^2 \quad \log 3.39794$

$\log 2.96115$

$\frac{1}{2}$ G = Gravity force 2580 pounds.

$\frac{1}{2}$ I = Inertia force 914 pounds.

PER CENT. OF FORCES BALANCED.

Forces to be balanced *Case I.—When point I, is below point G, the mean ordinate of the forces to be balanced = a = ordinate of gravity line.

Mean ordinates.

A = mean ord. of forces to be balanced = .860 in.

B = mean ord. of shaded area down stroke = .689 in.

C = mean ord. of shaded area up stroke = .972 in.

Per cent. of forces balanced.

Forces balanced down stroke $= \frac{A - B}{A} =$ 20.0 per ct.

Forces unbalanced up stroke $= \frac{A - C}{A} =$ 13.0 per ct.

Forces balanced mean — for 1 revolution = 4.0 per ct.

LOADS ON VALVE GEAR.

To be balanced. *Maximum loads.*	Pounds.
Max. load on ecc. bearing (from diagram) =	7,000
Max. pres. per □″ of proj. area of ecc. bearing =	41
Unbalanced.	
Max. load on ecc. bearing, (from diagram) =	6,306
Max. pres. per □″ of proj. area of ecc. bearing =	37
To be balanced. *Mean loads, (during a complete revolution.)*	
Mean load on ecc. bearing, A × 2 × 3,000 =	5,160
Mean pres. per □″ of proj. area of ecc. bearing =	30
Unbalanced.	
Mean load on ecc. bearing $\frac{(B + C}{2} \times 2 \times 3{,}000 =$	4,983
Mean pres. per □″ of proj. area of ecc. bearing =	29

* As the mean ordinates for both the Up and Down strokes are the same, it is necessary to calculate it only for the down stroke.

(5) That it is desirable to fit assistant cylinders on certain stationary engines.

THE LATEST TYPES OF ASSISTANT CYLINDERS.

In the latest types of assistant cylinders metallic rod packing has been substituted for the bushing, which is a distinct advantage, as all leakage is thereby prevented.

The assistant cylinder and the valve-chest bonnet are separate castings, the cylinder being bolted to the bonnet as shown. In case it is necessary to repair or overhaul the assistant cylinder, it can be unbolted and removed without interfering with the working of the engine, as there is no leakage past the balance piston rod.

BALANCE CYLINDER, U. S. S. *TENNESSEE.*

DATA SHEET.

Diam. of L.P. bal. piston = 10″ Area = 78.54
Diam. of L.P. bal. piston rod = $1\frac{3}{4}$″ Area = 2.41
Effective area = 76.13

L.P. valve gear:

Total gravity effect of valve gear to be balanced 5,805 lbs.
Gravity effect to be balanced by 1 balance cylinder

$$= \frac{G}{2} = 2{,}903.$$

Calculations and diagrams for *one* balance cylinder using $\frac{G}{2}$ and $\frac{I}{2}$.

Forces on diagrams are one-half of their real value.

BALANCE CYLINDER, U. S. S. *TENNESSEE.*

COMPUTATION SHEET.

Cylinder, L.P.
R.P.M., 125.

$I = 12{,}860$ $\qquad \frac{I}{2} = 6{,}430 = 2.15''$

Rec. pres. $= 38.5 \times 76.13 = 2{,}931 = .977''$

Forces to be balanced:

$B = 1.39$ $\quad H = 1.19$ $\quad \frac{1}{2}\,BH = .827$

$B_1 = 3.58$ $\quad H_1 = 3.10$ $\quad \frac{1}{2}\,B_1H_1 = 5.55$

$B + B_1 = 4.97$ $\quad$ Sum $= 6.377$

$$M = \frac{6.377}{4.97} = 1.28''$$

Unbalanced forces:

$b = 2.52$ $\quad h = 2.18$ $\quad \frac{1}{2}\,bh = 2.75$

$b_1 = 2.45$ $\quad h_1 = 2.13$ $\quad \frac{1}{2}\,b_1h_1 = 5.36$

$\quad$ Sum $= 8.11$

$$N = \frac{8.11}{4.97} = 1.63''$$

LOADS ON VALVE GEAR.

Maximum:

To be balanced $= H_1 \times 2s = 18{,}600$ lbs.

Unbalanced $= h \times 2s = 13{,}080$ lbs.

Mean:

To be balanced $= M \times 2s = 7{,}680$ lbs.

Unbalanced $= N \times 2s = 6{,}480$ lbs.

COMPUTING THE DATA.

Diam. of bal. piston (H.P., I.P. and L.P.) $= 9''$

Diam. of bal. piston rod $= 1\frac{7}{8}''$

Area of piston, top, $63.6\square''$

Area of piston, bottom, $60.9\square''$

Stroke of valve, $10''$

Crank arm of eccentric, $.416'$

Projected area of eccentric bearing surface, $173\square''$

WEIGHT OF MOVING PARTS OF VALVE GEAR.

H.P. valve gear.

	Pieces.	Weight.	Gravity effect.
Balance piston, . . .	1	65	65 lbs.
Valve, ...	1	1,130	1,130 lbs.
Valve stem, ...	1	400	400 lbs.
Link block, ...	1	130	130 lbs.
Links and pins, set, . .	1	430	215 lbs.
Suspension link, . . .	1	90	90 lbs.
Eccentric rod, ...	1	450	450 lbs.
Eccentric strap, . . .	1	845	845 lbs.
		Total =	3,325 lbs.

H.P. valve gear.—gravity effect of valve gear to be balanced = 3,330 lbs.

Note.—5 lbs. extra added.

I.P. and L.P. valve gear.

	Pieces.	Weight.	Gravity effect.
Balance piston, ...	2	130	130 lbs.
Valve, ...	2	2,250	2,250 lbs.
Valve stem, ...	2	360	360 lbs.
Link block, ...	1	130	130 lbs.
Links and pins, set, . .	1	430	215 lbs.
Suspension link, . . .	1	90	90 lbs.
Eccentric rod, ...	1	450	450 lbs.
Eccentric strap, . . .	1	845	845 lbs.
Crosshead, ...	1	685	685 lbs.
			Total = 5,155 lbs.

I.P. and L.P. valve gear.—Total gravity effect of valve gear to be balanced = 5,160 lbs.

Note.—5 lbs. extra added.

I.P. and L.P. valve gear.—Gravity effect of valve gear to be balanced *by 1 balance* cylinder (as used in computations) = 2,580 lbs.

CONSTANTS.

For converting pressures to the scale of the diagram, 6,000 lbs. = 1″.

Where

T = total pres. on balance piston;
p = pres. per sq. in.;
A = area of piston.

Inches to scale:

T = total pres. on piston;
S = scale*.

(*Scale, 6,000 lbs. = 1″)

Hence, inches to scale corresponding to pressure,

$$p = \frac{T}{S} = \frac{p \times A}{S} = p \times \frac{A}{S}.$$

For convenience let $\frac{A}{S}$ = constant.

Hence, inches to scale = $p \times$ constant.

CONSTANTS FOR ALL CYLINDERS.

All cylinders, same dimensions.

Scale 6,000 lbs. = 1″.

Top constant $= \frac{A}{S} = \frac{63.6}{6{,}000} = .0106$

Bottom constant $= \frac{A}{S} = \frac{60.9}{6{,}000} = .0101$

CONSTANTS FOR CALCULATING INERTIA.

I = maximum inertia, in pounds;
r = crank arm of eccentric, in feet;
N = revolutions per minute;
W = weight to be balanced, in pounds.

Formula for inertia:

$$I = .0003408\, rW \times N^2.$$

H.P. valve gear.

$$I = .0003408 \times .416 \times 3{,}330 \times N^2.$$

Let K = constant = $.0003408 \times r \times W$.

Then, $I = K \times N^2$.

.0003408,	log,	6.53250
$r = .416$	log,	9.61909
$W = 3{,}330$,	log,	3.52244
For H.P. valve gear, $K =$	log,	9.67403

I.P. and L.P. valve gears.

The I.P. and L.P. valve gears each have *two* balance cylinders.

The calculations will be made for *one* balance cylinder, and using ½ of the weight of valve gear to be balanced. *Hence, all pressures, gravity and inertia, are only ½ of their actual values for these valve gears.*

$w = 5{,}160 \qquad \frac{w}{2} = 2{,}580.$

$I = .0003408 \times .416 \times 2{,}580 \times N^2.$

or, $I = K \times N^2.$

.0003408,	log,	6.53250
$r = .416$,	log,	9.61909
$\frac{W}{2} = 2{,}580$,	log,	3.41162
For I.P. and L.P. valve gears, $K =$	log,	9.56321

DATA SUMMARY.

K for H.P., log, 9.67403
K for I.P. and L.P. log, 9.56321
G for H.P., 3,330 lbs. = 1.11 in.
G for I.P. and L.P., 5,160 lbs.
$\frac{G}{2}$ for I.P. and L.P., 2,580 lbs. = .86 in.

All cylinders.

Top, Constant $= \frac{A}{S} = .0106.$

Bottom, Constant $= \frac{A}{S} = .0101.$

Proj. area of ecc. bearing (all) = 173□″
H.P. has 1 asst. cylinder.
calculations for 1 asst. cyl.
forces represent actual values.
I.P. and L.P. have 2 asst. cyls. (each).
calculation for 1 asst. cyl.
forces represent ½ of actual values.

NOTES.

THE STABILITY OF SUBMARINES.*

By Sir William H. White, K.C.B., F.R.S.

The purpose of this paper is to place on record the results of calculations made to determine the conditions of stability of submarine vessels in varying circumstances which may occur on service. Accidents have happened to many submarines, and in some instances have been accompanied by loss of life. After investigating possible causes of accident, the author was convinced that one of the chief was the singular variation in stability and buoyancy produced by changes in the draught of water and the "trim" of submarines. He was led, therefore, to undertake the detailed calculations, of which the principal results are now stated and illustrated.

Either by accident or intention, submarines may reach considerable depths below the surface and be exposed to severe external fluid pressures. Ample structural strength must be provided to meet these pressures and to prevent deformation of the vessels. In order to fulfill this object with moderate weights of structures, submarines are made cigar-shaped, with circular or nearly circular cross sections. The cigar shape is usually somewhat disguised by light superstructures built above the upper surface of the hull proper, and carrying decks or platforms, which add to the comfort and convenience of the crews when the vessels are floating at the surface—in the "awash" condition—at their lightest draught. In that condition water is excluded from the spaces between the superstructures and the cigar-shaped hulls, and the buoyancy and stability are sensibly increased.

The other extreme condition at the surface is that when a

* Paper read before the Royal Society May 3, 1906.

submarine has been "trimmed" for diving, and floats with a very small portion of her hull above water. This is effected by admitting water ballast into tanks specially constructed for the purpose and of known capacity. The final adjustments of draught and trim during the process of trimming require great care. All openings into the interior are closed and secured in a watertight manner before trimming is commenced. Water is also allowed to enter the spaces between the superstructures and the cigar-shaped hull, and to remain in free communication with the surrounding water, so that the lightly-constructed superstructures may sustain no external pressure when the vessel is submerged.

Diving is accomplished by giving the submarine headway, and so manipulating horizontal rudders that the bow is depressed. The stream lines developed in the water by the onward motion produce downward pressures on the upper surface of the hull towards the bow; the vertical component of these pressures overcomes the vertical component of the rudder pressures and the small "reserve of buoyancy" which the submarine retains, and the vessel moves obliquely downwards until the desired depth below the surface is reached. The horizontal rudders must be then manipulated by a skilled steersman in such a manner that further motion (although really along an undulating course) is practically at a constant depth below the surface. When headway ceases, both rudder pressure and stream-line motions disappear, the small reserve of buoyancy reasserts itself, and the submarine rises to the surface.

This general statement may be illustrated by figures for an actual submarine, resting on official evidence given at the inquiry into the foundering of submarine *A*-8 at Plymouth last year.

In the awash condition, at the lightest draught of water, the reserve of buoyancy was about 13 tons (excluding the conning tower), the corresponding displacement exceeding 200 tons; so that the maximum reserve of buoyancy was about 6 per cent. of the displacement. The minimum reserve of buoyancy accepted for any class of warships at their deep-load draught

has been about 10 per cent. in low-freeboard American "monitors," many of which vessels foundered. For "breastwork monitors" in the Royal Navy the corresponding reserve was 30 per cent. of the load displacement; for high-freeboard warships and passenger steamers it is from 80 to 100 per cent.; for cargo steamers it varies from 25 to 40 per cent. The contrast between submarines at their lightest draught and other types of ships at their deepest draught, shown by these figures, indicates the acceptance of altogether exceptional conditions in submarines, and the necessity for their cautious management in the awash condition at the surface, when the apertures on the upper surfaces are kept open.

These apertures are closed and secured before the vessels are trimmed for diving by admitting water ballast. In the diving condition the reserve of buoyancy is extremely small. For submarine *A*-8 it is said to have been 800 pounds, the corresponding displacement being about 220 tons. In other submarines of about the same displacement the reserve of buoyancy in the diving condition has been only 300 pounds to 400 pounds. Consequently there is a necessity for extreme care in the final stages of trimming.

The cigar shape of the hulls involves very rapid changes in the areas and moments of inertia of the planes of flotation, as the draught of water is increased in passing from the awash to the diving condition; the stability is greatly reduced, and every member of the crew has to remain in his station. No weights must be allowed to shift. All the conditions, in fact, differ from those which prevail in ships of ordinary form as they pass from the extreme light draught to the deep load, for in such ships the outlines of transverse sections approximate to the vertical, except near the bow and stern, over the range between these extreme conditions, and the areas and moments of inertia of planes of flotation do not vary greatly.

These general statements may be illustrated by the comparison of a small cruiser of ordinary form with a submarine. The cruiser is about 260 feet long at the water line, 37 feet broad, and 14 feet 6 inches mean-load draught, the correspond-

ing displacement being about 2,000 tons. The submarine has an extreme length of 150 feet, is 12.2 feet in extreme breadth, and has a displacement of 300 tons in the diving condition. In the light (awash) condition the submarine draws about 18 inches less water than in the diving condition, and has a displacement of about 284 tons. When awash the length at the water line is 94 feet, and breadth, extreme, 8.2 feet; when in the diving condition the corresponding measurements are 41 feet length and 3.6 feet breadth. These figures differ widely from the length of 150 feet over all and 12.2 feet maximum breadth. In the cruiser, within the corresponding range of draught (18 inches), there is practically no change in length and breadth, extreme, at water line, and these dimensions are practically identical with the extreme dimensions of the vessels. Fig. 1 illustrates the contrast between the cigar shape and

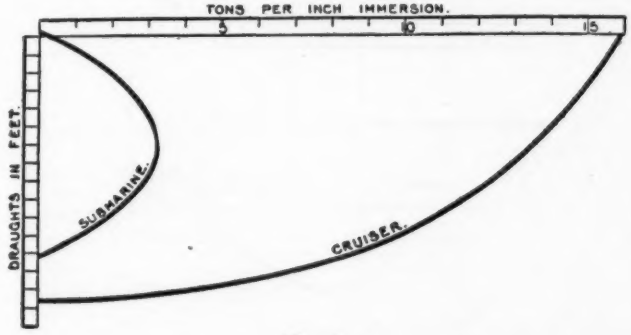


Fig. 1.

the ordinary ship shape. Horizontal measurements to the curves on that diagram, at any draught of water, measure the area of the corresponding plane of flotation, and the number of tons required to immerse the vessel 1 inch. It is obvious that the small area of the plane of flotation at the lightest draught, its rapid diminution as the draught is increased, and the critical condition when trimmed for diving, all render possible the establishment of vertical dipping oscillations in submarines by comparatively trifling disturbances in the water surface surrounding them.

Fig. 2 shows the metacentric diagrams for transverse inclinations of the two vessels, constructed in the usual manner. M M shows the *locus* of the metacenter of the cruiser, and B B that of the center of buoyancy as the draught of water varies. The curves *m, m, m* and *b, b, b* show the corresponding *loci* for the submarine. The intercept between these curves on any vertical ordinate represents the height of the metacenter above the center of buoyancy at the corresponding draught of water. For the submarine in the awash condition this height is 0.32 foot, and in the diving condition it is 0.01 foot. Stability

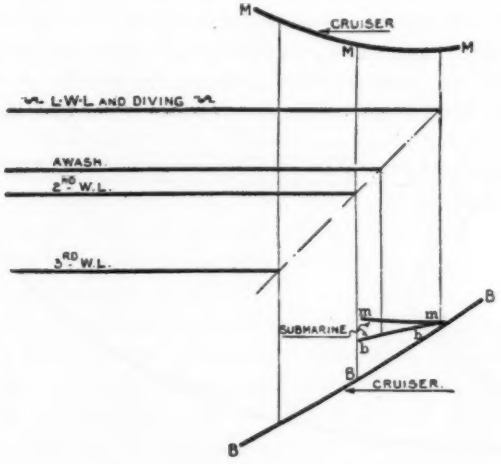


Fig. 2.

is obtained by disposing the weights so that the center of gravity of the vessel and its contents lies below the axis; and in some existing submarines in their diving condition the vertical distance between the axis and the center of gravity, or metacentric height, is said to be about 9 inches. When submerged this measure of stability, of course, applies to inclinations from the vertical in any direction.

For the cruiser the height of the metacenter above the center of buoyancy is 7.7 feet at the load-water line, and 8.6 feet for

the water line (18 inches below the load), corresponding to the awash condition of the submarine. The center of gravity of the cruiser is about 2 feet below the metacenter and 5¾ feet above the center of buoyancy in the load condition; the metacentric *locus* is nearly horizontal from the load to the light condition, and the center of gravity rises a few inches as coals and stores are consumed.

Fig. 3 shows the metacentric diagrams for longitudinal inclinations. For the submarine awash the metacenter is 37 feet

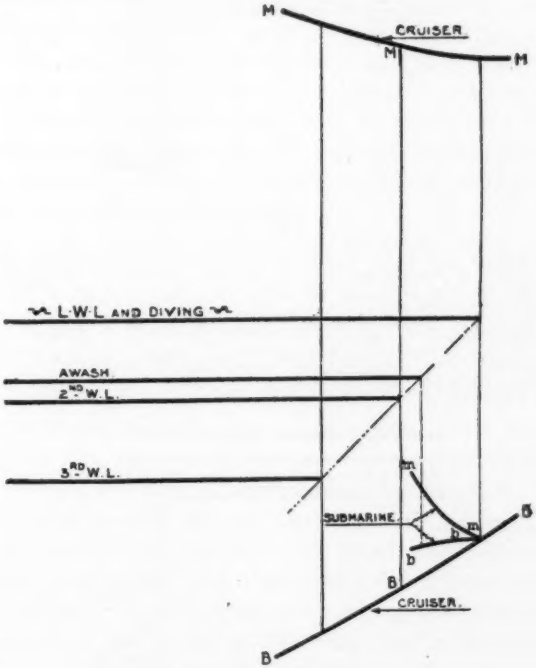


Fig. 3.

above the center of buoyancy; in the diving condition it is only 1.25 feet above. For the cruiser floating at a water line 18 inches below the load draught, the height is 352 feet; at the load draught it is 328 feet. Expressed in terms of the length over all, the heights of metacenters above centers of

buoyancy are 0.25 and 0.0083 times the length respectively for the awash and diving conditions, as against 1.35 and 1.26 times the length for the cruiser at corresponding draughts. These figures indicate the relatively small longitudinal stability of the submarine, and the necessity for avoiding any movements of weights when the vessel is in the diving condition or submerged.

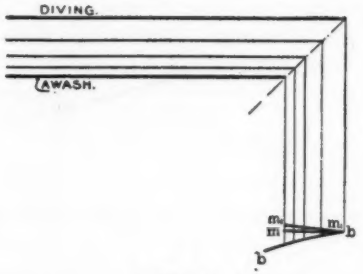


Fig. 4.

Reference has been made above to the effect upon stability produced by the addition of superstructures. Fig. 4 illustrates this effect for transverse inclinations, m, m being the metacentric *locus* without superstructure, and m_1, m_1 the *locus* with superstructure closed and water excluded from spaces between

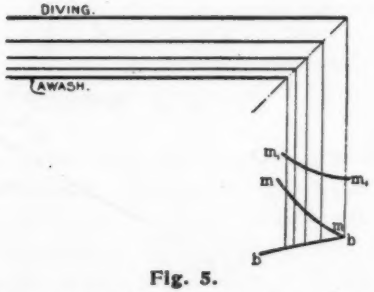


Fig. 5.

it and the cigar-shaped hull. In the awash condition the height of the transverse metacenter above the center of buoyancy is increased about one-sixth by the superstructure. The effect of the superstructure upon longitudinal stability is much more marked, as will be seen from Fig. 5. In the awash con-

dition, closing the superstructure increases the height of metacenter above the center of buoyancy by fully 50 per cent.

It will be obvious from these diagrams that the maintenance of the full reserve of buoyancy is essential to the safety of a submarine when proceeding at maximum speed at the surface. In the case of *A*-8, owing to special circumstances, this condition was not fulfilled, and the vessel proceeded at full speed on the surface with her ballast tanks partly filled with water, and with only 6 tons reserve of buoyancy, as against the maximum reserve of 13 tons. In consequence of this deeper draught the longitudinal metacentric height was reduced from 12 feet to 8½ feet, and the power of resisting changes in longitudinal trim was correspondingly diminished. Since that accident took place, definite orders have been given by the Admiralty that the maximum reserve of buoyancy shall always be secured before submarines are driven at full speed on the surface. The precaution is obviously necessary.

When a submarine is in the diving condition, with all apertures closed and crew stationed, the metacentric height (as above stated) is very small, and the trim may be sensibly and rapidly disturbed by small external forces. Consequently very moderate angles of helm given to the horizontal rudders by the operator will produce sensible changes of trim; and as the pressures on the rudders vary as the square of the speed of the vessel, increase in speed, with consequent increase in rudder pressures, demands greater skill and precaution on the part of the helmsman. A very small amount of trim "by the bow," in association with moderate speed when submerged, will bring a submarine to a considerable depth below the surface in a very short time. Experience proves that with trained and disciplined operators at the helm, and with moderate speeds such as have been accepted hitherto, submarines can be worked at fairly constant depths below the surface. On the other hand, many cases have occurred where submarines have reached considerable depths, and have touched bottom in consequence of slight accidents or failure in control. These considerations point to the conclusion that much higher speeds

than have been obtained hitherto when submerged must be accompanied by greatly increased risk; and it may be questioned if the gain in offensive power obtained by increased speed justifies the change in these circumstances. For large submarines it is universally agreed that automatic appliances for regulating depth below the surface are not to be trusted, although they are successful in locomotive torpedoes.

Close approximations can be made to the pressures developed on the horizontal rudders of a submarine moving at a given speed, and to the corresponding changes of trim produced in the vessel. Similar approximations cannot be made at present to the pressures and inclining moments consequent on the stream-line motions in the water surrounding a submarine when she moves ahead. This matter can only be dealt with by direct experiment on models and submarines. In the course of the inquiry into the foundering of *A*-8 this conclusion was universally accepted. Differences of opinion existed as to the primary cause of that accident. It was obvious that the deeper draught, the lessened stability, and the open hatch all conduced to the disaster; but experienced witnesses asserted that they were not of opinion that the vessel could have been made to dive suddenly, as she did, if she possessed as much as 6 tons reserve of buoyancy. Others equally experienced entertained the opinion that this was the real cause of the accident. After a careful analysis of the evidence the author was convinced that the latter opinion was correct. It was stated at the time that the Admiralty proposed to have experiments made at their experimental tank, and on actual submarines, in order to settle this difference of opinion. Up to the present time no results of such Admiralty experiments have been published; if they have been made, this silence is much to be regretted on scientific grounds, and no reason is seen for refusing the information. It has been stated authoritatively that experiments of the kind have been made on models of submarines at the Experimental Establishment of the United States Navy at Washington, and that the results have confirmed the opinion expressed by the author.

In connection with the inquiry into the loss of *A*-8 it was made known that her commanding officer recognized the fact that lessened stability must accompany deeper immersion, and that he trimmed the vessel 4 degrees by the stern (lifting the bow about 4 feet or 5 feet) in the belief that this change would make the vessel less liable to be driven under water by the stream-line action on the bow.

In considering all the circumstances the author was consequently led to investigate the variations in stability accompanying changes of trim in submarines, and to compare them with corresponding changes in other ships. The technical term "trim" here used means the difference in draught of water at the bow and stern; it has no relation to "trimming" for diving. It was obvious, of course, that the cigar shape must introduce variations in stability with change of trim much greater than those which would occur in vessels of ordinary form, and it was known that in ordinary vessels the changes of trim which occur in service are not of practical importance. Figs. 6 and 7 give the results obtained for the submarine awash and for

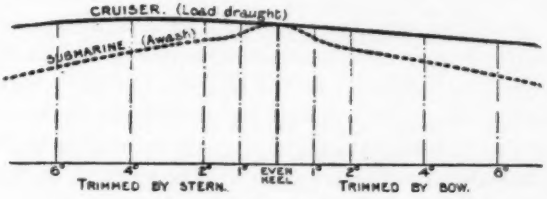


Fig. 6.

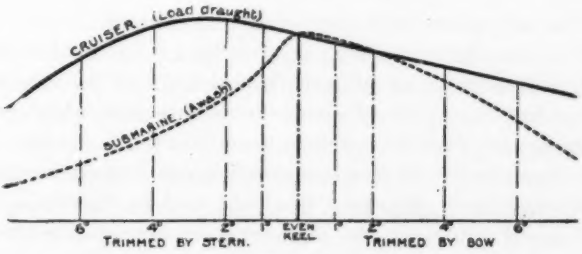


Fig. 7.

the cruiser at load draught, when changes of trim take place by the bow and stern, up to 6 degrees from the "even keel" condition. In order to compare the two types more closely, the heights of metacenters above centers of buoyancy for the even-keel condition are treated as unity in both cases, although they differ widely, as above stated. Ordinates to the curves at any angle of trim measure the relative heights of the corresponding metacenter above the center of buoyancy. Fig. 6 shows these heights for transverse inclinations, and Fig. 7 those for longitudinal inclinations. In both cases the effect of superstructures is omitted. Longitudinal stability is more important, and the results may be briefly summarized. Taking 4 degrees trim by the stern, the height of the metacenter above the center of buoyancy in the submarine is only 45 per cent. of the height when the vessel is on an even keel. For the cruiser the corresponding figure is 100 per cent.—that is, there is practically no change in longitudinal stability within the limit of trim mentioned. If the superstructure came into play in the submarine, the percentage of the metacentric height at 4 degrees by the stern to the height on even keel would exceed 50 per cent. It will be seen, therefore, that for a cigar-shaped vessel departures from even keel are accompanied by serious decrease in longitudinal stability, and it may be doubted whether the depressing effect of the stream-line motions at the bow would be reduced to an equal extent, if at all, by raising the bow to the extent done in the case of *A*-8. The latter point, however, is determinable only by direct experiment.

Figs. 8 to 10 represent three conditions of draught and trim for the submarine dealt with in the calculations.

The foregoing statements lead to the conclusion that in the design of submarines the calculations for stability require to be worked out by naval architects to an extent which is not necessary for ships of ordinary form, and that each departure from precedent must be most closely scrutinized and exhaustively considered. It is true, no doubt, that for the diving and submerged conditions, the essential point is to deal accurately

with questions of weight and position of the center of gravity, since stability in all directions when submerged depends upon the relative positions of the centers of gravity and buoyancy, and moderate "metacentric heights" have to be accepted. On the other hand, it is certain that equal attention should be directed to the conditions of stability in the awash condition, and in the stages of immersion between it and the diving condition. Submarine design is not a task to be entrusted to amateurs or imperfectly informed persons. Skilled naval architects alone should undertake the work, and the results of their investigations should be put into the form of simple practical rules for the guidance of officers and men. From the nature of

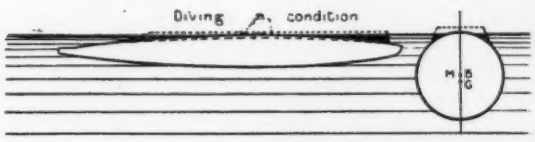


Fig. 8.

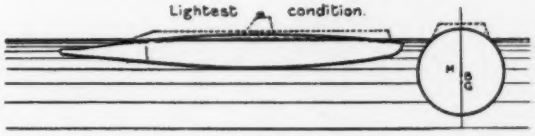


Fig. 9.

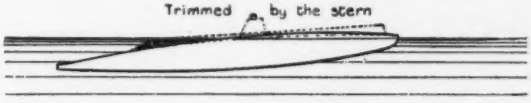


Fig. 10.

the case—in consequence of the singular forms of the vessels, the small reserves of buoyancy, and the exceptional variations in stability which must be accepted in order to obtain the power of rapid submergence—considerable risks must be taken. It is therefore the duty of all concerned to give all possible assistance to officers and crews in the form of information and instructions based on thorough investigation and experiment. —"Engineering," London.

TABLE OF DIMENSIONS OF PRINCIPAL ATLANTIC LINERS OF MARITIME NATIONS.

Nationality	Britain.	Britain.	Germany.	France.	United States.
Name of ship	*Lusitania.*	*Campania* and *Lucania.*	*Kaiser Wilhelm II.*	*La Provence.*	*St. Paul* and *St. Louis.*
Builders	John Brown & Co., limited.	Fairfield Company, Glasgow.	Vulcan Company, of Stettin.	S. des Chantiers et Ateliers de Saint Nazaire.	Cramp, of Phila.
Owners	Cunard Company.	Cunard Company.	North German Lloyd.	Cie General Transatlantique.	American.
When built	1905-7.	1893.	1902	1905-6.	1895.
Length over all	785′	622′	706′ 6″	624′ 8″	554.2′
Length between perpendiculars	760′	600′	...	597′ 1″	535.8′
Breadth, molded	88′	65′ 3″	72′	64 7½″	63′
Depth, molded	60′ 6″	41′ 6″	52′ 6″	41 8″	42′
Gross tonnage, tons	32,500	12,500	20,000	...	11,629
Draught, feet	33	25	29	26′ 9″	26
Displacement, tons	38,000	18,000	26,000	19,160	16,000
Number of passengers, first	550	600	775	442	320
second	500	400	343	132	200
third.	1,300	700	770	808	800
Type of engine	Turbine.	Five-cylinder triple-expansion.	Four sets, four cylinder quadruple-expansion.	Four-cylinder triple-expansion.	Six-cylinder quadruple-expansion.
Number of cranks in each engine	...	3	6	4	4
Diameter of cylinders, inches	...	Two of 37, one of 79, and two of 98.	Four of 37.4, four of 49 2, four of 74.8, four of 112.2.	Two of 47.2, two of 76.3, four of 88.18.	Two of 28, one of 55, one of 77 and two of 77.
Stroke, inches	...	69	70.86	66.9	60
Number and type of boilers	Twenty-five cylindrical.	Twelve double and one single ended.	Twelve double and seven single ended.	Twenty-one cylindrical.	Six double and four single ended.
Number of furnaces	192	102	124	84	64
Steam pressure, pounds	200	165	225	200	200
Total heating surface, square feet	160,000	82,000	107,643	58,342	40,320
grate area, square feet	3,950	2,630	3,121	1,571	1,144
Draught	Howden's.	Open stokehold.	Open stokehold.	Howden's.	Howden's.
Total indicated horsepower	68,000	30,000	38,000 to 40,000	30,000	18,000
Speed, knots	25†	22.01	23½	22.05*	21.08

† Designed speed. * On trial.

TURBINE-DRIVEN GERMAN TORPEDO-BOAT DESTROYER *S-125*. (*VIETH.*)

(Translated from "Marine Rundschau."

The construction of six similar destroyers for the German Government, numbered *S*-120 to *S*-125, inclusive, gave an opportunity to institute comparisons between the efficiency of reciprocating engines and of turbines, as *S*-125 was a turbine-driven vessel, while the others had reciprocating engines. The six vessels were built by the well-known torpedo establishment of F. Schichau, Elbing, except that the Parsons turbines for *S*-125 were furnished by Brown, Boveri & Co., Baden. The hulls were, as far as possible, identical in construction, but the engine room of *S*-125 was much crowded on account of the head room being limited to that allowed in the other vessels.

The turbine installation of *S*-125 comprises a main and a cruising engine, the first composed of one high- and two low-pressure turbines, and the latter of one high- and one low-pressure turbine. The main turbines operate three shafts, each with one screw, the H.P. in the center and the L.P. in the wings, while the cruising turbines are placed forward of the main L.P. turbines and are coupled on the same shafts as the latter. For backing, there are two high-pressure turbines placed, as is customary, at the after end of the main L.P. turbines and within the casings of the latter.

The construction of the turbines does not differ from others of the Parsons type, except that the main bearings are provided with water circulation.

The condensers are about one-fifth larger than the condensers of the boats with reciprocating engines, and the circulating pumps of twice the power. There are three air pumps, two wet and one dry, the former of the Weir type, making about 50 double strokes per minute at full power, and the latter driven by a reciprocating engine running about 250 revolutions per minute. The displacement per minute of the three pumps is about 38 per cent. more than that of the direct-connected pumps of the other vessels.

The method of operation of the turbines is as follows:

For speed above 17 knots, only the three main turbines are used.

For speed up to 14 knots, both cruising turbines are coupled up with the main L.P. turbines, and, under this condition, steam is admitted to the H.P. cruising turbine, passes to the L.P. cruising turbine, thence to the main H.P., and from that to both main L.P. turbines and to the condensers.

For speed up to about 15.5 knots, the cruising turbines are coupled up in parallel with the main turbines. Live steam is admitted into the H.P. and L.P. cruising turbines at the same time, but as the H.P. exhausts to the L.P. cruising turbine, the amount of this exhaust is limited and regulated by an automatic cut-out valve; when the pressure gets too high, the H.P. is automatically cut out and runs free. Under this condition, a speed of about 17 knots can be obtained.

On the main H.P. turbine there is a passover valve by means of which live steam can be admitted to the 10th stage, and in this way an increase in power can be obtained, provided the boilers are capable of supplying the steam, but this valve was not used during the trials of *S*-125.

When working to signals it is customary to use only the main L.P. and the backing turbines, special pipes being provided for this purpose.

During the trials of *S*-125 three designs of screws were tried, and in general it was found that the one of greatest pitch and greatest projected area gave the greatest speed, but also required the greatest expenditure of power; while the screws of less pitch and area gave less speed and also a lower coal consumption; from which it was concluded that, in order to reach a good speed without abnormal expenditure of coal, it will be necessary to make a compromise in the type of screw. The pitch ratio of the screws tried varied from 1.39 to 1.29. The ratio of disc to projected area of the three screws was 2.6, while for the reciprocating engines it was 3.4.

TRIALS.

The contract required that, when fully equipped and loaded with sufficient coal for a radius of action of 2,000 miles at 12 knots, the vessel should make a coal-consumption trial of 14 hours' duration at a speed of 12 knots; and that the coal consumption as thus determined should be the basis for fixing the quantity of coal to be carried on succeeding trials. (*S*-120 to *S*-124 required only 315 I.H.P. for this speed; their maximum power was 6,500 I.H.P.)

As a result of this coal-consumption trial it was determined that *S*-125 should carry 99 tons of coal on trial, or 46 tons more than the others. To this must be added a few tons for increased weight of turbines and of hull, making the trial displacement 447 tons as against 396 for the reciprocating-engine vessels.

On this trial *S*-125 burned 1,309 pounds of coal per hour, or 87 per cent. more than the vessels with reciprocating engines. The revolutions of the turbines were 312 per minute.

With increased power the coal consumption dropped. At 17 knots, at which speed the others required 918 I.H.P. and 1,878 pounds of coal per hour, the coal consumption of *S*-125 was 2,874 pounds, or 53 per cent. more than for the reciprocating engines. The turbines ran at 498 revolutions per minute.

The nearer the speed approached the maximum the greater was the increase in economy.

On the 3-hour forced-draft trial, for which the contract required 27 knots, the speed was 26.62, while the best speed of the sister vessels was 28.27 knots. (See page 669 of last number of the JOURNAL.) The revolutions per minute were 784.

On a subsequent trial under similar conditions, except that the coal carried was the same as that carried by the vessels with reciprocating engines, the speed was about 28 knots with 818 revolutions of the turbines.

Another requirement of the contract was that the vessels should run with the backing turbines at a speed of 15 knots.

(*S*-120 to *S*-124 required 1,315 I.H.P. for this.) Under this condition the revolutions of the wing screws were 533 and of the center screw, running idle, 360 per minute. The small number of stages in the backing turbines was very unfavorable to economy, and it was found necessary to use full forced-draft boiler power to make the speed, resulting in excessive pressure in the condensers.

There was also a trial in the Baltic during heavy weather, with wind 8 to 10, during which the speed, determined from the revolutions of the screw, was 15 knots. The vessel proved to be a good sea boat. The center screw was never out of water, and the wing ones but seldom, and then only partially so, their revolutions increasing only about 50 per minute.

Tests were also made in handling the turbines. The time required to carry out an order was about the same as with reciprocating engines, but the time necessary to bring the vessel up to a given speed, or to stop her was a little greater.

With the same rudder area the turning circle was greater, but with a slight increase in area it was the same as for the reciprocating engined vessels.

GENERAL OBSERVATIONS ON THE INSTALLATION.

The results of the trials demonstrated that, without material increase in weight and space, better economy can not be hoped for at speeds below the maximum, though it has not been fully demonstrated that experiments with different screws might not accomplish something.

The weight and space required are greater than for reciprocating engines.

The engine room is very crowded, rendering any work of repair or overhauling extremely difficult and tedious.

The trials showed that the turbines were not as free from derangement as had been hoped. The small clearance between the outer edges of the blades and the casing leads to trouble if there be unequal expansion between the rotor and casing, or there has been unusual wear of the bearings on account of a lack of oil, causing that bending and breaking of the blades

called by the builders "blade salad." Such an accident, due to the causes above recited, occurred to the cruising turbines.

Small clearances are indispensable to economy, and with efficient oil service an appreciable wear of the bearings is not to be anticipated.

In preparing the turbines for use they must be as carefully warmed up as a reciprocating engine, in order that unequal expansion of the casings and rotors may be avoided. This is especially true of the cruising turbines whose clearances are the smallest.

Fewer men are required in the engine room, the oiling arrangements are more satisfactory, and the consumption of oil much less than with reciprocating engines.

The backing turbines are too wasteful of steam; they should have a number of stages such that one boiler will be sufficient in backing.

There was an absence of vibration with the turbine installation.

COAL-TAR PAINT.

The article on "Coal-Tar Paint" in our last (May) issue, by Civil Engineer A. C. Cunningham, U. S. Navy, Member, has been extensively copied by the leading engineering papers of the country. In the "Engineering News" correspondence of August 16, 1906, Mr. Frank P. Cheesman, apparently a paint manufacturer, takes occasion to criticise coal-tar paint in an abstract way, treating the subject as if the paint were raw coal tar. He states that gas companies once protected their gas holders with coal tar but "are fully persuaded that it is a poor article to use." In the same correspondence Mr. Alfred E. Forstall, lately Secretary of the American Gas-Light Association, states that while raw coal tar makes an unsightly coating, especially after repeated applications, he is personally familiar with cases where it has preserved a priming coat of red lead on gas tanks for twenty years. Mr. Forstall further states that, in his experience, coal tar

boiled until free from water, neutralized with a little quick lime and thinned with light tar oils, makes a very satisfactory paint, the best results being had when applied hot on a warm surface.

It will be noted that in the mixture proposed by Mr. Cunningham the Portland cement, being basic and having an affinity for water, takes the place of both boiling and quick lime and its composition admits of its being applied hot where possible or desirable.

SHIPS.

UNITED STATES.

New Hampshire—Launch of.—The battleship *New Hampshire* was successfully launched on June 30, 1906, from the yards of the New York Shipbuilding Company, Camden, N. J.

The general dimensions and features of the *New Hampshire* are as follows: Length over all, 456 feet 4 inches; on load-water line, 450 feet; breadth, extreme, at load-water line, 76 feet 10 inches; displacement on trial, 16,000 tons; mean draught bottom of keel, 24 feet 6 inches; speed, 18 knots; total bunker capacity, 2,314 tons. The main battery is to consist of four 12-inch breech-loading rifles; eight 8-inch breech-loading rifles; twelve 7-inch breech-loading rifles and four submerged torpedo tubes. The second battery is to consist of twenty 3-inch, 14-pounder, rapid-firing guns; twelve 3-pounder semi-automatic guns; four 1-pounder semi-automatic guns; two 3-inch field pieces, two machine guns of .30 caliber and two automatic guns of .30 caliber. The 2-inch guns are to be mounted in pairs in two electrically-controlled, balanced, elliptical turrets on the center line, one forward and one aft, each with an arc of fire of about 270 degrees. The 8-inch guns are to be mounted in pairs in four electrically-controlled, balanced, elliptical turrets, two on each beam at each end of the superstructure. The 7-inch guns are to be mounted in broadside, protected by 7-inch armor, the forward and after guns being arranged to fire directly ahead and astern, respectively. The hull is to be protected at the water line by a complete belt of armor, 9 feet 3 inches wide, having a uniform thickness of 9 inches for about 280 feet amidships, gradually decreasing to 4 inches at the stem and stern. A complete protective deck will extend from stem to stern.

The propelling engines will be right-and-left, placed in watertight compartments and separated by a middle-line bulkhead. The engines will be of the vertical, inverted-cylinder, direct-acting, triple-expansion type, each with a high-pressure cylinder 32½ inches, an intermediate-pressure cylinder 53 inches, and two low-pressure cylinders 61 inches diameter, the stroke of all pistons being 48 inches. The indicated horsepower of propelling engines will be 16,500 when making about 120 revolutions per minute, and with a steam pressure of 250 pounds at the high pressure cylinder. There will be two condensers, each with a cooling surface of about 10,375 square feet, measured on the outside of the tubes, the water passing through the tubes. There will be twelve water-tube boilers, placed in six water-tight compartments, with three smoke pipes. The total grate surface is to be 1,100 square feet, and the heating surface at least 46,750 square feet. Provision is to be made for forced draft by blowers discharging into airtight firerooms, there being two blowers in each boiler compartment.

Minnesota—Dock Trials.—Satisfactory dock trials of this vessel were held at the works of the contractors, the Newport News Shipbuilding & Dry Dock Company, August 14-17. The *Minnesota* is a first-class battleship, a general description of which will be found in Vol. XVII, pp. 540-543.

Milwaukee—Official Trials.—The official trials of this vessel were held in Santa Barbara Channel, California, July 25. A full description of the vessel and her trials will appear in the November issue of the JOURNAL.

Nebraska—Official Trials.—This vessel underwent her official trials in Puget Sound, Washington, July 16. A complete description of the vessel and her trials will be given in the next issue of the JOURNAL.

South Carolina—Contract for.—The contract for this vessel was awarded July 21, 1906, to the Wm. Cramp & Sons Ship and Engine Building Company, Philadelphia, Pa. The contract price is $3,540,000, and the vessel is to be completed in 41 months.

The *South Carolina* is a first-class battleship with the following general characteristics:

Principal Dimensions.—Length, on load-water line, 450 feet; length between perpendiculars, 450 feet; length over all, 452 feet 9 inches; breadth, extreme, to outside of armor, 80 feet 2⅝ inches; trial displacement, about 16,000 tons; trial draught, to bottom of keel, 24 feet 6 inches; speed, 18½ knots.

Main Battery.—Eight 12-inch breech-loading rifles; two 21-inch submerged torpedo tubes.

Secondary Battery.—Twenty-two 3-inch, 14-pounder, rapid-fire guns of 50 calibers in length; two 3-pounder semi-automatic guns for saluting; eight 1-pounder semi-automatic guns; two 3-inch field pieces; two machine guns of 0.30 caliber; two automatic guns of 0.30 caliber.

Propelling Machinery.—The propelling engines will be right-and-left, placed in watertight compartments and separated by a middle-line bulkhead. The engines will be of the vertical, inverted-cylinder, direct-acting, triple-expansion type.

The general particulars of engines will be as follows:

Diameter of high-pressure cylinder, inches	32
intermediate-pressure cylinder, inches	52
forward low-pressure cylinder, inches	72
after low-pressure cylinder, inches	72
Stroke of all pistons, inches	48
Indicated horsepower of both main engines	16,500
Revolutions per minute	125
Steam pressure, at high-pressure steam chest, pounds	265

Each main condenser will have a cooling surface of about 9,500 square feet, measured on the outside of the tubes, the water passing through the tubes.

The main circulating pumps will be of the centrifugal type, one for each main condenser. For each main condenser there will be an independent air pump of the double, vertical, single-acting type.

There will be an auxiliary condenser, of about 550 square feet cooling surface, in each engine room, connecting with all the auxiliary machinery.

There will be two auxiliary condensers of about 1,050 square feet cooling surface, each, for the dynamo room, for the exclusive use of the dynamo engines. If steam turbines are used, the dynamo-condenser cooling surface will be as approved.

Each auxiliary and dynamo condenser will have an air and circulating pump.

There will be twelve water-tube boilers, with superheaters, placed in three watertight compartments.

Total grate surface, square feet	1,050
Boiler-heating surface, square feet	42,075
Superheater heating surface, square feet	4,675
Total heating surface, square feet	46,750

The forced-draft system will consist of blowers discharging into airtight firerooms, there being four blowers for each boiler compartment.

Michigan—Contract for.—This battleship is a sister ship to the *South Carolina.* The contract for her construction was awarded July 20, 1906, to the New York Shipbuilding Company, Camden, N. J. The contract price is $3,585,000, and the vessel is to be completed in 40 months.

Summary of Naval Construction.—

BATTLESHIPS.

Name of vessel.	Building at	1906: Per cent. of com. July 1.	Aug. 1.
Virginia	Newport News S. B. Co.	100.00	
Nebraska	Moran Bros. Co.	95.16	96.0
Georgia	Bath Iron Works	97.47	98.43
Connecticut	Navy Yard, N. Y.	97.41	97.6
Vermont	Fore River S. B. Co.	88.5	90.5
Kansas	New York S. B. Co.	85.2	85.8
Minnesota	Newport News S. B. Co.	91.44	93.68
Mississippi	Wm. Cramp & Sons.	59.86	60.83
Idaho	Wm. Cramp & Sons.	58.07	59.0
New Hampshire	New York S. B. Co.	48.0	50.3
South Carolina	Wm. Cramp & Sons.	.....	0.
Michigan	New York S. B. Co.	.....	0.

Name of vessel.	Building at	1906: Per cent. of com. July 1.	Aug. 1.
ARMORED CRUISERS.			
California	Union Iron Works	94.0	94.6
South Dakota	Union Iron Works	91.8	92.4
Tennessee	Wm. Cramp & Sons	99.75	100.0
Washington	New York S. B. Co	99.0	100.0
North Carolina	Newport News S. B. Co	50.5	53.87
Montana	Newport News S. B. Co	44.72	47.66
PROTECTED CRUISERS.			
St. Louis	Neafie & Levy S. & E. B. Co.	98.77	99.22
Milwaukee	Union Iron Works	96.1	97.9
TRAINING SHIPS.			
Cumberland	Navy Yard, Boston	95.0	95.0
Intrepid	Navy Yard, Mare Island	97.5	97.5
SCOUT CRUISERS.			
Chester	Bath Iron Works	36.88	40.8
Birmingham	Fore River S. B. Co	38.4	41.8
Salem	Fore River S. B. Co	39.3	42.6
SUBMARINE TORPEDO BOATS.			
Submarine T. B. *No. 9*	Fore River S. B. Co	75.9	81.7
Submarine T. B. *No. 10*	Fore River S. B. Co	62.0	69.3
Submarine T. B. *No. 11*	Fore River S. B. Co	72.3	81.3
Submarine T. B. *No. 12*	Fore River S. B. Co	59.0	65.1

AUSTRIA-HUNGARY.

The Austrian Government proposes to reorganize the Navy, and in the next estimates will propose the construction of three battleships and several fast cruisers. The Minister of Marine emphasized the necessity of establishing a naval base at Sebenico, at the mouth of the river Kerka. The biggest warships owned by Austria-Hungary do not exceed 10,000 tons displacement, and therefore they are of no use in the naval warfare of today; thus, the naval authorities will undertake on the first opportunity to replace the old warships *Tegethoff*, dating from 1878, and the *Erzherzog Rudolf* and the *Erzherzogin Stephanie*, both built in 1887, by three new battleships of not less than 14,000 tons each.

Sankt Georg—Trials of.—The following is the performance of this vessel on her official trials recently completed:

At the twelve hours' coal-consumption trial, with the engines developing 6,322 I.H.P. and making 103.03 revolutions, the mean speed was 17.54 knots, with a consumption of 1.06 kg. (2.33 pounds) per I.H.P. per hour, and 143 kg. (314 pounds) per square meter of grate surface, eight boilers being in use; at the second twelve hours' trial, with the engines developing 9,206 I.H.P., and making 117 revolutions, the mean speed was 19.6 knots, with a coal consumption of 0.99 kg. (2.17 pounds) per I.H.P. per hour, and 130.7 kg. (287.5 pounds) per square meter of grate surface; at the four-hours' full-speed trial under natural draft, the engines developed 13,095 I.H.P., making 130.1 revolutions, the mean speed of 21.3 knots was maintained; and at the two-hours' full-speed trial under forced draft, with the engines developing 15,271 I.H.P., and making 134.9 revolutions, a speed of 22.01 knots was maintained on the measured mile. The displacement of the *Sankt Georg* at the time of her trials was 7,420 tons, complete for sea, except provisions, of which half the full stowage only was on board. For a general description of the *Sankt Georg* see Vol. XVIII, p. 300.—"Mittheilungen aus dem Gebiete des Seewesens" and "Marine Rundschau."

Ulan.—The first of five torpedo-boat destroyers of the Huszar type building at Trieste, had her machinery trials on the 23d of May, and realized a speed of 29.2 knots. The speed of the *Huszar,* tried a month later, was 28.54 knots.

These destroyers are of the following dimensions: Length, 219.8 feet; beam, 20.37; displacement, 390 tons (metric). Their engines are of 6,000 I.H.P. The battery comprises one 70- and seven 47-mm. guns.

ENGLAND.

H. M. S. Agamemnon, first of the *Lord Nelson* class of British battleships to be put into the water, was launched on the Clyde, on June 23d, from the naval construction works of Messrs. Wm. Beardmore & Co.

The keel of the *Agamemnon* was laid on May 15, 1905, and she was therefore launched in thirteen months from the beginning of operations. She is in a fairly advanced stage, having in place a considerable part of her armor and the whole of the fifteen Yarrow water-tube boilers which constitute the steam-generating plant. The launching weight of the ship is 7,000 tons. This constructional performance has frequently been exceeded, but the work has been prosecuted at Dalmuir under disadvantageous circumstances, as the shops have been in course of completion while the work of constructing the hull has been in progress.

The armament is as follows: Four 12-inch 45-caliber guns in twin barbettes; eight 9.2-inch 50-caliber guns in twin barbettes; two 9.2-inch 50-caliber guns in single barbettes; thirteen 12-pounder 12-cwt. quick-firing guns; twenty-two 3-pounder quick-firing guns; two 12-pounder 8-cwt. quick-firing boat and field guns; two Maxims; four broadside submerged torpedo tubes; one stern submerged torpedo tube.

The secondary guns are located on a flying deck 13 feet above the upper deck, extending for about one-third the length of the ship, and occupying only about one-half the width. This is a new feature, or, rather, is a revival of the old feature. The forward ends have sheltered screens, and the conning tower is at the forward end of this flying deck. It is of 12-inch armor, and, as in recent ships, has no opening in its circumference at the rear, entrance being through a trap door from the chart room above. This trap door is, like the roof of the conning tower, of armor, so that there is no chance of splinters finding their way into the conning tower, as in earlier ships. The boats are carried about 7 feet above the flying deck. The location of the machine guns on this flying deck gives them a much better command against torpedo attack owing to their greater elevation.

The armor over the machinery and other vital parts is 12 inches in thickness, tapering to 6 inches at the bow and 4 inches at the stern; above this the armor is 8 inches up to the upper deck. Forward and aft there are the usual bulkheads,

which are also of 8-inch armor. Within them are located the barbettes, which are for the most part of 12-inch armor, while the gun mountings of the 9.2-inch guns are protected by armored barbettes, with very heavy hoods over the mechanism. The magazines for the 12-inch guns are located forward, and here also the greater part of the ammunition for the 9.2-inch guns will be stored; but for quick service in action, shell and powder chambers are provided between the boiler and the machinery spaces, provision being made not only for adequate ventilation, but also for cooling the chambers.

The machinery of the *Agamemnon* is being constructed by Messrs. R. and W. Hawthorn, Leslie & Co., of Newcastle-on-Tyne.

The propelling engines are of the standard British Admiralty type, having four cylinders working on the triple-expansion system. The high-pressure cylinders in each set of engines are 32¾ inches in diameter, the intermediate 53¾ inches; the low-pressure cylinders 60 inches in diameter, the stroke in each case being 48 inches. The engines take steam at an initial pressure of 250 pounds, the working pressure in the boilers being 275 pounds. The propellers are 15 feet in diameter, the pitch being 19 feet, and the surface of the four blades 90 feet square. It is anticipated that with the engines working at 120 revolutions these propellers will give the vessel a speed of 18 knots when she is loaded to her mean draught. The power of the machinery at 120 revolutions will be 16,750 horsepower. The ship is 410 feet long between perpendiculars and 79 feet 6 inches breadth, molded.

Minotaur.—This armored cruiser, launched June 6 at Devonport, is one of three which are being built in the Royal dockyards. The sister ship *Defence* is under construction at Portsmouth, and the *Shannon* is being built at Chatham. The principal dimensions of the *Minotaur* are: Length, 490 feet; beam, 74 feet 6 inches; mean draught, 26 feet, and displacement, 14,000 tons. Her engines of 27,000 I.H.P., which are being made by Messrs. Harland & Wolff, of Belfast, are of the triple-expansion type. The speed

for which the vessel is designed in 23 knots. The boilers, twenty-five in number, will be of the Babcock & Wilcox water-tube type, arranged in five separate boiler rooms, and loaded to 275 pounds per square inch. Amidships the *Minotaur* will carry a belt of 6-inch Krupp armor, tapering forward to 4 inches and aft to 3 inches. The barbettes will be protected by 7-inch and the conning tower by 10-inch armor. The armament will comprise four 9.2-inch guns in two barbettes, ten 7.5-inch guns in ten barbettes, 16 12-pounder 18-cwt. quick-firing guns, two 12-pounder 8-cwt. quick-firing guns, and five Maxims. There will also be four submerged torpedo tubes amidships, and one at the stern. The launching weight of the ship was about 6,600 tons.

H. M. S. Gadfly, the first of the new coastal destroyers to be completed in connection with last year's naval programme, was, on May 28, 1906, successfully launched from the Chiswick Yard of Messrs. John I. Thornycroft & Co., Ltd. The dimensions of this vessel are: Length, 168 feet; beam, 17 feet 6 inches; draught, 5 feet 11 inches; and the contract speed, 26 knots. The *Gadfly* is one of five sister vessels now building by Messrs. Thornycroft, fitted with turbine machinery of Parsons type and Thornycroft water-tube boilers. The armament will consist of two 12-pounder quick-firing guns and three torpedo tubes.

Loss of Torpedo Boat No. 56.—*No.* 56 was one of five torpedo boats specially commissioned at Malta for service in the Suez Canal in connection with the boundary dispute with Turkey. At 5 P. M. on June 17th, she left in tow of the *Arrogant,* on her return to Malta, and at 1:30 A. M. the following morning, when off Damietta, she capsized, one first-class petty officer, an able seaman, an engine-room artificer, a leading stoker, and three stokers being unfortunately drowned. There was some sea on, apparently, at the time, and if the boat got into a cross sea the disaster would probably be due to this.

Stranding of the Montagu.—In a dense fog about 2 A. M. on the morning of May 30th the first-class battleship *Montagu,* belonging to the Channel Fleet, struck on the Shutters,

a dangerous ledge of rocks on the southwest end of Lundy Island. The *Montagu* was employed in testing some new appliances designed to prevent wireless messages being tapped. She seems to have anchored in a fog in the roads off the north end of Lundy on the previous morning, but to have weighed again in the evening with the intention of going down Channel. The vessel has been abandoned as a total loss.

Loss of Torpedo Boat No. 84.—During the operations off Malta on the night of April 17th, torpedo boat *No.* 84 was run into by the *Ardent* and sunk.

Steam Trials of H. M. Armored Cruiser Cochrane.—The armored cruiser *Cochrane,* the third of the cruisers of the *Duke of Edinburgh* class, has just completed her trials, and the results, so far as speed is concerned, are quite satisfactory. This vessel, which was constructed and engined by the Fairfield Shipbuilding and Engineering Company, Limited, Govan, N. B., belongs to the first class of cruisers designed by Sir Philip Watts since he went to the Admiralty as Director of Naval Construction, and has a very considerable increase in gun power, as compared with preceding ships, having six 9.2-inch and four 7.5-inch guns, with twenty-six quick-firers; whereas in the earlier ships there were fitted many 6-inch weapons, which experience has since thoroughly established are not efficient in long-range fighting. This increase in armament necessarily involved an augmentation in all the dimensions of the ship, the vessels having 30 feet more length than the improved County class and 2,700 tons more displacement. The length of the *Cochrane* is 480 feet, the beam 73 feet 6 inches, and the draught 27 feet, with a displacement of 13,550 tons. Part of the increased weight is due to improved armor protection. It was found desirable to reduce the proportion of weight allowed for machinery, consequently the engine power per unit of displacement was not increased in the same ratio as the displacement itself. In other words, the improved County class have 1.93 indicated horsepower per ton displacement, whereas the new vessels have 1.73 indicated horsepower per ton. It was anticipated, however, that the increased length

and modifications in the design of propellers would give a speed of 22.33 knots. As a matter of fact the trials, so far, of the three ships of the class have shown that they are as fast as any cruisers in the Navy. The *Cochrane*, although she only developed 149 horsepower in excess of the designed power, or 23,649 indicated horsepower, attained a speed of 23,292 knots over a measured course, and this has been about the average for the vessels of this class so far tried. They are thus a nautical mile per hour in excess of the designed speed. The model has thus been particularly satisfactory, and the propellers very efficient.

On the first trial of 30 hours' duration, which was run at one-fifth of the full power, the engines at 82.75 revolutions developed 4,911 indicated horsepower, which gave the vessel a speed of 14.3 knots. On the 30-hours' trial, at about 70 per cent. of the total power, the engines, running at 122.35 revolutions, developed 16,080 horsepower, giving the ship a speed of 21.37 knots, according to runs made over a measured course. At full power the engines were run at 135.2 revolutions and developed 23,649 indicated horsepower. The designed revolutions were 135, to give 23,500 indicated horsepower, so that the actual result closely approximates to that anticipated. The air pressure in the stokeholds on this full-power trial was 0.8 inch in the case of the cylindrical boilers, and 0.64 inch where water-tube boilers were installed.

A feature of the trials was the coal and steam consumption. Owing to the stoppage of the Penrikyber Colliery, it was not possible to get a supply of fuel which is deemed most satisfactory for steaming purposes, and a lower grade of coal had to be resorted to. Thus the coal-consumption results of the trials are not so favorable as they might have been, averaging about 2 pounds per indicated horsepower.

The steam consumption is, however, exceedingly favorable. For the main engines it averages 15 pounds per horsepower, having been 15.3 pounds on the low-power trial and 14.4 pounds on the 30-hours' trial. This is considerably better than in the case of the two preceding ships. On the eight-hours'

Table I.—Results of Steam Trials of H. M. S. "Cochrane."

	June 24 and 25, 1906. 30 hours at 4,700 indicated horsepower.	June 30, 1906. 30 hours at 15,700 indicated horsepower.	July 5, 1906. 8 hours at 23,500 indicated horsepower.
Date of trial			
Duration and nature of trial			
Draught of water, feet and inches	*Forward.* 26–7 | *Aft.* 27–5	*Forward.* 26–6½ | *Aft.* 27–6	*Forward.* 26–6 | *Aft.* 27–6
Steam in boilers, pounds per square inch	188.4	194	189
	Starboard. | *Port*	*Starboard.* | *Port.*	*Starboard.* | *Port.*
Vacuum, inches	26.8 | 27.6	26.8 | 27.4	26.4 | 26.3
Revolutions per minute	83.0 | 82.5	122.3 | 122.4	135.2 | 135.2
Mean pressure, high, pounds	33.6 | 32.3	64.8 | 63.1	81.8 | 79.2
intermediate, pounds	12.2 | 12.1	27.0 | 27.9	33.6 | 33.6
low, fore, pounds	4.6 | 4.8	11.5 | 11.8	17.7 | 17.7
low, aft, pounds	4.5 | 4.7	12.2 | 12.3	17.9 | 18.1
Mean indicated horsepower, high	886 | 843	2,504 | 2,439	3,487 | 3,378
intermediate	813 | 802	2,621 | 2,719	3,610 | 3,609
low, fore	391 | 401	1,404 | 1,437	2,367 | 2,369
low, aft	382 | 393	1,472 | 1,484	2,403 | 2,426
Mean total indicated horsepower	2,472 | 2,439	8,001 | 8,079	11,867 | 11,782
Mean collective indicated horsepower	4,911	16,080	23,649
air pressure, pounds	Nil	{ water-tube, 0.36 cylindrical, 0.4	water-tube, 0.64 cylindrical, 0.8
speed of ship, knots	14.3 by bearings.	{ 21.37 on measured mile.	23.292 on measured mile.
coal consumption per indicated horsepower per hour, pounds	2.0	2.04	1.99
Total loss of water, tons	12.6	55.5	13.4
Water used by main engines per indicated horsepower per hour, pounds	15.3	14.4	16.4
Water used by auxiliary engines per indicated horsepower per hour, pounds	3.3	2.2	16.4

full-power trial the water used by both main and auxiliary engines was only 16.4 pounds, as compared with 19 pounds in the case of the *Duke of Edinburgh* and *Black Prince.*

The propelling machinery includes two sets of four-cylinder triple-expansion engines, arranged in separate watertight compartments. The diameters of the cylinders are—high-pressure, 43½ inches; intermediate, 69 inches; and two low-pressure, 77 inches; the stroke in all cases being 42 inches. The machinery generally is exactly similar to that of the *Duke of Edinburgh* and *Black Prince,* having been made from corresponding gauges, jigs, and templates to ensure interchangeability.

The boiler installation, however, differs. In the *Cochrane* the six cylindrical boilers in No. 4 boiler room work under the closed-stokehole system, the air being supplied by four motor-driven 35-inch Sirocco fans. Each of these boilers is 13 feet 6 inches in external diameter and 10 feet 6 inches long, and has plain tubes 2½ inches in external diameter, 8 L.S.G. thick, and stay tubes 2½ inches in external diameter by ¼-inch thick. In each boiler there are three Deighton's corrugated furnaces, 3 feet 3 inches in internal diameter. The grate surface is 60.9 square feet, and the heating surface 1,904.5 square feet. The shell plating is 1 5/16 inches thick.

There are nineteen water-tube boilers of the Yarrow type, having a total grate surface of 971 square feet, and a total heating surface of 55,534 square feet. These are arranged in three separate boiler rooms—seven boilers in the forward or No. 1 boiler room, and six each in Nos. 2 and 3 rooms. Each boiler has a steam drum 4 feet 2 inches in diameter, 888 generating tubes 1¾ inches in external diameter by 5/32 inch thick. The water pockets are of the form usually fitted in the Yarrow boiler, and in addition to the generating tubes are connected to the steam drum by four down-comers 4 inches in internal diameter by 7/32 inch thick. The portion of the steam drum into which the tubes are expanded is 1 9/10 inches thick, while the remaining portion is 9/16 inch thick. The boiler casings were zincked by the hot process, and are fitted with the usual sight doors, lagging, etc.

Each boiler room is fitted with separate main and auxiliary feed pumps, by Messrs. G. & J. Weir, Cathcart, either of which is capable of feeding the boilers when working at full power. Nos. 1, 2 and 3 boiler rooms are fitted with See's ash ejectors, the water supply to which is delivered by two duplex pumps, by Messrs. Clarke, Chapman & Co. No. 4 boiler room is fitted with an ash tube of extra large diameter, and a motor-driven ash hoist. Each boiler room is provided with the necessary soot-cleaning valves for compressed air at a pressure of 50 pounds per square inch, which is supplied by a three-cylinder single-stage compressor by Messrs. H. Watson & Sons, Newcastle. The Yarrow boilers are also worked under the closed-stokehold system, the air being supplied by motor-driven fans.

Steam Trials of H. M. Armored Cruiser Natal. — His Majesty's armored cruiser *Natal,* constructed by Messrs. Vickers Sons & Maxim, Ltd., Barrow-in-Furness, has completed all the steam trials specified in the contract, and the results are, in every instance, very satisfactory, the efficiency in some respects exceeding that in other ships of the class. The *Natal* belongs to the *Duke of Edinburgh* class, and is 480 feet long between perpendiculars, 73 feet 6 inches beam, and at 27 feet draught displaces 13,550 tons. She carries 2,000 tons of coal, in addition to a large quantity of oil fuel in the double bottom. She is particularly well protected, for the most part with 6-inch armor, and has six 9.2-inch and four 7.5-inch guns, mounted singly above the upper deck in barbettes of 6-inch armor.

The machinery of all six vessels of the class has been built to gauges, templates and jigs to secure standardization as far as it is practically possible. The boiler installation includes six cylindrical boilers, 13 feet 6 inches in diameter by 9 feet 7 inches long, with a collective heating surface of 11,310 square feet and a grate area of 327 square feet, and nineteen Yarrow water-tube boilers, with an aggregate of 55,534 square feet of heating surface and 971 square feet of grate area. The following figures summarize the results of the boiler performance:

Table I.—STEAM TRIALS OF H. M. S. "NATAL."

	8 hours' full power.		30 hrs' consumption trial at 15,700 indicated horsepower.		30 hrs' consumption trial at 4,700 indicated horsepower.	
Duration of trial						
	Forward.	*Aft.*	*Forward.*	*Aft.*	*Forward.*	*Aft.*
Draught of water, feet and inches	26-5¼	27-6⅛	26-7½	27-5½	26-9	27-2
Speed of ship (knots per hour) from measured mile	23.344		21.35		14.8	
Steam pressure in boilers, pounds per square inch	195.0		183.9		164.5	
Air pressure in stokehold, Yarrow boilers, inch	0.8		0.3		...	
cylinderical boilers, inch	1.0		0.9		...	
	Starboard.	*Port.*	*Starboard.*	*Port.*	*Starboard.*	*Port.*
Vacuum in condensers, inches	27.0	27.0	27.8	27.6	26.1	26.8
Revolutions per minute	137.3	136.5	123.6	123.9	83.66	82.83
Mean pressure in receivers, high, pounds	185.0	184.0	166.9	168.9	114.4	119.5
intermediate, pounds	71.7	72.6	54.8	59.4	29.9	30.5
low, pounds	19.9	19.0	9.07	8.7	1.21	2.3
Mean pressure in cylinders, high, pounds	77.8	78.1	61.89	61.81	25.87	28.97
intermediate, pounds	35.0	35.2	26.48	26.25	13.57	13.43
low, forward, pounds	16.7	17.3	11.86	12.13	5.35	5.29
low, aft, pounds	17.4	16.8	12.39	12.57	5.18	4.97
Mean indicated horsepower, high	3,367	3,361	2,408	2,414	682	756
intermediate	3,818	3,812	2,594	2,581	899	882
low, forward	2,270	2,336	1,398	1,486	442	433
low, aft	2,360	2,268	1,513	1,543	429	390
Total	11,815	11,777	7,913	8,024	2,452	2,461
Grand total	23,592		15,937		4,913	
Consumption of coal per indicated horsepower per hour, pounds	2.01		1.8		2.03	

	At one-fifth power.	At seven-tenths power.	At full power.
Heating surface per indicated horsepower of main engines, square feet	7.15	4.2	2.82
Coal consumed per square foot of grate surface, pounds	16.2	22.4	39.25
Indicated horsepower per square foot of grate area	8.0	12.0	17.6
Indicated horsepower per ton of machinery	...	7.3	10.75

The official results of the three trials are reproduced in Table I; the speed is particularly satisfactory. At 15,937 indicated horsepower the rate maintained throughout the thirty-hours' trial was 21.35 knots. This was deduced in the usual way by taking on the measured mile the revolutions required to give this rate; this was corroborated further by the speed over long-distance runs during the progress of the trial. Although the designed power was only exceeded by 92 indicated horsepower, being 23,592 indicated horsepower, the speed attained—23,344 knots—was fully one nautical mile in excess of that anticipated by design, which suggests a high efficiency in the propeller and the minimum of friction in the machinery. The results prove that, with the coal and oil-fuel supply carried, the radius of action at 14½ knots is 6,700 miles; at 20½ knots, 3,500 miles; and at full speed, 2,300 miles. The period embraced by the trials, it may be added, was considerably protracted owing to fog. In connection with the thirty-hours' run at one-fifth power, three unsuccessful attempts, of considerably over thirty hours' duration collectively, were successively made, the ship being at sea for fifty hours. But in accordance with Admiralty practice the thirty-hours' trial had to be an uninterrupted run. The *Natal* will now be completed for commission.

Steam Trials of H. M. Battleship Africa.—The steam trials of the battleship *Africa,* built at the Chatham Dockyard and engined by Messrs. John Brown & Co., Limited, Clydebank, were of more than usual interest, as the whole of the bearings

were fitted with a system of forced lubrication, partially tried in one or two earlier ships, but here exclusively adopted. She is 425 feet long between perpendiculars, 78 feet beam, and at 26 feet 9 inches draught displaces 16,350 tons. The engines are of the four-cylinder compound type, the high-pressure cylinder being 38 inches in diameter, the intermediate 60 inches, and the two low-pressure cylinders 67 inches; the stroke being 48 inches. There are four condensers, two for each set of

OFFICIAL STEAM TRIALS OF H. M. BATTLESHIP, "AFRICA."

Date of trial	May 31 and June 1 and 2, 1906.	June 3, 1906.
Nature of trial, I.H.P.	30 hours at 12,600	8 hours at 18,000
Draught of water, feet and inches	*Forward.* 26–4 | *Aft.* 27–1½	*Forward.* 26–4 | *Aft.* 27–1½
Speed of ship, knots	17.547 (mean of six runs on measured course.)	18.953 (mean of six runs on measured course.)
Steam pressure in boilers, pounds	194	199
Air pressure in stokeholds, inches	Nil	0.49
	Starboard. | *Port.*	*Starboard.* | *Port.*
Vacuum in condensers, inches	26.8 | 27.1	26.6 | 27.2
Revolutions per minute	115.7 | 115.8	129.1 | 128.8
Mean pressure in receivers :		
High, pounds	166.0 | 167.0	171.0 | 173.0
Intermediate, pounds	49.0 | 47.0	63.0 | 62.0
Low, pounds	7.3 | 5.5	19.0 | 16.0
Mean pressure in cylinders :		
High, pounds	63.4 | 70.4	75.9 | 82.3
Intermediate, pounds	27.6 | 28.6	31.5 | 33.1
Low forward, pounds	11.1 | 10.0	18.16 | 15.7
Low aft, pounds	10.4 | 10.1	17.45 | 15.5
Indicated horsepower, high	2,020 | 2,243	2,699 | 2,902
intermediate	2,191 | 2,269	2,788 | 2,923
low, forward	1,097 | 996	2,003 | 1,729
low, aft	1,029 | 1,002	1,920 | 1,707
Total	6,337 | 6,510	9,410 | 9,261
Grand total	12,847	18,671
Coal per indicated horsepower per hour, pounds	1.79	1.87
Total actual loss of water, tons	18.9	10.19

engines, and the total cooling surface is 21,500 square feet. As in the case of several ships ordered three or four years ago, the steam generating plant is a combination of cylindrical and water-tube boilers, there being three of the former, with 235 square feet of grate area and 8,328 square feet of heating surface, and eighteen Babcock & Wilcox boilers, with 1,250 square

feet of grate area and 40,068 square feet of heating surface. The working steam pressure is 210 pounds.

The trials which terminated on June 3, 1906, passed off most satisfactorily; the table above gives the results of the thirty-hours' trial at 75 per cent. of the power and of the eight-hours' run at full power—the two most crucial tests. As will be seen at a glance, the Clydebank firm have repeated their previous successes with warships. On the thirty-hours' trial the mean power was 12,847 indicated horsepower with the engines making 115.75 revolutions, while the consumption was 1.79 pounds per unit of power per hour. The loss of water on the trial was 18.9 tons for the whole thirty hours, which is less than 25 per cent. of that allowed by the contract. Six runs were made over a long, deep-sea course, and the speed was found to be 17.547 knots. On the full-power trial the speed was 18.953 knots, as compared with the 18½ knots anticipated in design. The power was 18,671 indicated horsepower, or 671 indicated horsepower more than that required by the contract.

The coal consumption was 1.87 pounds, the loss of water—10.19 tons—again comparing most favorably with that allowed by the specification. The vessel is now to be prepared for commission.

FRANCE.

New Ships.—The new first-class battleship *République,* having received her armament, and being in all other respects ready, has been placed in the roadstead at Brest, and having received her reduced complement, is now to carry out her steam and other trials; she is the first of the battleships of the 1900 Programme to be completed.

Orders have been received at Brest and Lorient for the laying down of the two new 18,000-ton battleships, *A*-16 and *A*-15, respectively; the remaining four are to be built in private yards. All six ships are to be completed within four years. Their characteristics, which have now been finally approved by the Minister of Marine, are as follows: Length, 475 feet 7 inches; beam, 84 feet; draught, 27 feet 6 inches, with a dis-

placement of 18,000 tons. The armament will consist of four 12-inch guns in two turrets, one forward and one aft; twelve 9.4-inch guns in pairs in turrets, three on each broadside; sixteen 12- and eight 3-pounders, with two submerged torpedo tubes. The engines are to develop 22,500 I.H.P., giving a speed of 19 knots. Details of the armor have not yet been published. The ships are to be called the *Voltaire, Diderot, Condorcet, Vergniaud, Mirabeau* and *Danton.*

The Armored Cruiser Jules Ferry—Trials of.—The *Jules Ferry* is the second cruiser of the *Leon Gambetta* class, and has the following particulars: a length on the load-water line of 515 feet 1 inch; a beam of 70 feet 1 inch, and a mean draught of 26 feet 10 inches. The corresponding displacement is 12,606 tons. The coal capacity at normal draught is 1,400 tons, and the total bunker capacity is 2,100 tons, besides 100 tons of liquid fuel.

There are three sets of engines, four-cylinder, triple-expansion type, operated by steam from du Temple-Guyot boilers (small torpedo-tube boiler type).

The interesting features of the trials were:

Two hours with four boilers, 8,500 horsepower, and 16 knots.

Six hours with all boilers, 17,000 horsepower and 19 knots.

Twenty-four hours with all boilers, 16,508 horsepower and 18.5 knots.

Four hours with full power, 29,200 horsepower and 22.48 knots.

The coal consumption on six-hour trial was 650 grams (1,433 pounds) per horsepower-hour; on the 24-hour trial it was 744 grams (1.64 pounds); and on the four-hour trial 800 grams (1.764 pounds) per unit of power per hour.

The cruiser was run on the measured mile off Brest, where there is much deeper water than off Cherbourg. The *Leon Gambetta* having made off Brest an average speed of 23.46 knots, it was hoped the *Jules Ferry* could get the same speed.

The battery consists of four 7.7-inch guns in turrets, fore and aft, and sixteen 6.4-inch guns, twelve in turrets on the

spar-deck and four in casemates; besides twenty-two 3-pounder guns; there are also five torpedo tubes.

GERMANY.

Roon, cruiser of the *Gorck* type, had her coal-consumption trials in May, the result being a speed of 13.64 knots on 3,800 I.H.P., and a coal consumption of 1.97 pounds per I.H.P.

There are building at the Germania Works, Kiel, 6 destroyers—*G*-132 to *G*-137—five of which, with reciprocating engines, are of 486 tons displacement, and the sixth, *G*-137, with turbines, of 572 tons. The latter will have four boilers instead of three, and is expected to make 30 knots instead of 28.5 for the others. *G*-135 will mount three 3.46-inch and two 2-inch guns, and the others two 3.46-inch and four 2-inch.

Four battleships, the *Kaiser,* the *Deutschland,* the *Preussen* and the *Friedrich der Grosse,* built immediately after the foundation of the German empire, have been ordered to be stricken from the list of effective vessels of the German Navy. In the course of eighteen months five battleships, four cruisers and three gunboats have been dropped from the list of effectives.

GREECE.

Naval Notes.—The first of two 31-knot torpedo boat destroyers which Messrs. Yarrow & Company are building for the Greek Government was launched at Poplar on July 23d. These two vessels are similar in type to the destroyers built by this firm for the Japanese Government, having a normal displacement of about 360 tons. Their length is 220 feet; their beam, 20 feet 6 inches; and the mean normal draught about 5 feet 9 inches. They are fitted with twin screws and four-cylinder triple-expansion engines of 6,500 indicated horse-power, balanced in accordance with the Yarrow, Schlick and Tweedy principle. The cylinder diameters are: High-pressure, 20½ inches; intermediate, 31½ inches; and the two low-pressures, 34 inches; and the piston stroke is 18 inches. They are designed to develop the full power when running at 390 revo-

lutions per minute. The boilers, four in number, are of the Yarrow water-tube "express" type, and have a combined heating surface of 13,000 square feet, with a grate area of 220 square feet. They are designed for a working pressure of 250 pounds per square inch. The total weight of the machinery is about 138 tons. The armament of the Greek vessels is more powerful than that of the Japanese destroyers, there being two 12-pounder and four 6-pounder quick-firing guns, and two torpedo tubes for 18-inch torpedoes.

MERCHANT SHIPS.

Lusitania.—The new Cunard Line turbine steamship *Lusitania,* the largest vessel in the world, was successfully launched from the yard of John Brown & Co., Clydebank, Scotland, on Thursday, June 7. She was christened by Dowager Lady Inverclyde. The *Lusitania* is the first of two giant liners building for the Cunard Line to be launched. Her sister ship, the *Mauritania,* will be put overboard in a month or six weeks hence.

The exact length of the *Lusitania* is 785 feet over all (the general idea has prevailed that the ship was to be 800 feet over all), and the length between perpendiculars 760 feet. This makes her 60 feet longer than the White Star liner *Baltic,* which is 725 feet in length, and which has heretofore been the world's greatest steamship.

The *Lusitania* is 88 feet breadth, molded, and 60 feet 6 inches depth, molded. The gross tonnage is 32,500, and at 33 feet draught will displace 38,000 tons. She will have accommodations for 2,400 passengers and will carry 700 officers and assistants.

There are four turbines, turning four distinct screws. The two outside turbines are for steaming ahead only. The two turbines on the inner shafts will steam ahead and astern also. The turbines are all designed on the compound principle. Those on the outside shafts take steam direct from the boilers, which work at 200 pounds per square inch, and in turn exhaust into the two low-pressure turbines on the inner shafts. The low-pressure engines weigh about 430 tons, the rotor having a diameter of 15 feet 8 inches, and the blades, which, as in all turbines, increase in length toward the low-pressure end, attain a maximum length of 22 inches, so that the maximum diameter of the rotor with its blades is 19 feet 4 inches. The

clearance between the blades and the proximate surface—*i. e.*, between the rotor blades and the casing, and between the fixed blades and the rotor—varies, from high- to low-pressure ends, from ½ inch to ¼ inch. The peripheral speed of the rotor under normal conditions of working will attain a maximum of 142 feet per second. Some idea can thus be formed of the enormous importance of that precision and reliability which has been aimed at, alike in design and workmanship, by Mr. Thomas Bell, the engineering director at the works, who has evolved many most important improvements. Without this fineness of fit there might be loss through leakage; with inequalities there might be friction, resulting in the blades being stripped. This precision is the key to success in turbine work, and the fact has in this instance been thoroughly recognized. Again, the proportions of the casing, to resist circumferential stress without undue weight, involved careful consideration. Cast iron has been used for the casings, and forged steel for the rotors, and the whole of this heavy work has been done at the Sheffield works of John Brown & Co. The line shafting, which was also forged at the company's Sheffield works, is 22 inches in diameter in the case of the low-pressure systems, and the rotor spindle mounted on this shafting has a diameter of 33 inches.

The wing propellers are a considerable distance ahead of the inside propellers, but, owing to the great beam of the ship and the very fine run, the blades of these propellers do not project beyond the fore-and-aft beam line of the ship. All the shafts are enclosed within the framing, which is bossed out for the purpose. To improve the stream lines, and to facilitate maneuvering, the "deadwood" of the ship aft is cut away, forming an arch; the propellers work in free water, but the blades do not overlap. Abaft of the propellers the keel is again sloped downwards to form a support for the rudder. The stern frame, weighing 50 tons, and the brackets, of 30 tons, hydraulically riveted to it, are of cast steel. In the stern frame is incorporated the foundation for the bearings of the rudder stock, with an extension for the rear crosshead of the

steering gear. There are separate brackets, with heavy flanges, built into the ship for the wing propeller-shaft bearings. The rudder, which is of the balanced type, has a framing of steel, and weighs about 65 tons; the rudder head is about 25½ inches in diameter.

The steam-generating plant includes twenty-five boilers, having in all 192 furnaces. The total grate area is just under 4,000 square feet, and the heating surface 160,000 square feet, so that there are 40 square feet of heating surface per unit of grate area. The ratio of legend power to heating surface is 1 : 2.35. This is, perhaps. a little less than in some of the earlier Atlantic liners: in the *Kaiser Wilhelm II* the ratio is 1 : 2.75; but there the open stokehold system of draft was adopted. In the *Deutschland,* where the Howden system is also applied, the ratio is 1 : 2.37; in the latest French ship it is rather less. The boilers are arranged in four compartments, each with its uptake, and thus there are four large funnels. The funnels are oval on plan, 24 feet being the greater diameter, and 155 feet the height from the keel.

The 20-knot Atlantic liner of twenty years ago had a displacement of 10,500 tons; the 22-knot ship of ten years ago 18,000 tons; the 23-knot ship of five years ago 26,000 tons; and now to reach 25 knots it has been found necessary to raise the displacement weight to 38,000 tons.

The dominant factor in this large increase of displacement is the great addition to engine power for a comparatively slight increase in speed. Thus twenty years ago the ratio of power to displacement for a 20-knot speed was 1.35 to 1; for 22 knots ten years ago, 1.66 to 1; and today, for 25 knots, it is 1.8 to 1. Thus for an advance in speed in twenty years equal to 25 per cent., the power has increased from 14,000 to 68,000, or nearly 400 per cent. It is true that with the improved economy the coal consumption per unit of power is considerably less—25 per cent.; but notwithstanding the less time taken on the voyage from Liverpool to New York, owing to increased speed, the total tonnage of coal used has gone up from 1,900 to 5,000 tons. Since the first Cunarder was

built, sixty-five years ago, the speed has been multiplied only threefold, but the power has had to be increased nearly a hundredfold. Owing, however, to the higher economy, and the shorter time taken on the voyage—about four and a half days as compared with fifteen days—the coal consumption for the voyage is only about nine times as great.

A significant feature of the dimensions of the *Lusitania* is the beam of 88 feet. This width was determined as the result of a careful series of model experiments, and resulted in a very fine model, the block coefficient of fineness being very low. The length of the *Lusitania* is only 8.63 times the beam, whereas in the case of the *Campania,* of ten years ago, the length is 9.2 times the beam, and in some of the high-speed German ships it is 9.5 times the beam. The molded depth is greater than in earlier ships, but, in addition to this, there is, as compared with the Atlantic liner of ten years ago, an additional superstructure deck, so that there are eight decks in all. This has been rendered possible by the great beam, which enables a satisfactory measure of stability to be preserved. It affords a considerable addition to passenger accommodation above the water level. Twenty years ago the draught was seldom over 23 feet, but this was increased to 29 feet in the case of the present record-breaking Atlantic liners. In the *Lusitania,* however, it is hoped that there will be sufficient depth of water to enable the ship to leave port with a 33-foot draught. This considerably increases the wetted surface, and consequently the skin resistance; but with deeper immersion the propellers work more efficiently.

The *Lusitania* will probably make her builders' trials in June, 1907, and should be ready for service sometime during the summer of 1907.—"The Nautical Gazette."

Kaiserin Auguste Victoria.—This, the largest vessel of the Hamburg-American Line, was built by the Vulcan Works at Stettin, and is slightly larger than the sister ship *Amerika*, built last year by Harland & Wolff, Belfast, for the same company. The keel was laid August 4, 1904. She was

launched August 29, 1905, and in May, 1906, made her first voyage.

Her principal dimensions are:

Length over all, feet	698.83
between perpendiculars, feet	675.87
Breadth, extreme, feet	77.00
Depth to upper deck, feet	53.74
Draught, mean, feet	32.97
Displacement at above draught, tons	35,500
Speed, knots	17
Tonnage, gross register	24,581
Bunker capacity, tons	3,500

She is schooner rigged, with four pole masts, between the two middle of which are the smoke pipes.

The engines are twin-screw, quadruple-expansion, of the following dimensions.

Working pressure, pounds	213.00
Cylinder, H.P., diameter, inches	36.22
1st I.P., diameter, inches	53.15
2d I.P., diameter, inches	75.59
L.P., diameter, inches	106.69
stroke, inches	64.96
Crank shaft, diameter, inches	20.67
Bilge pump (1), single-cylinder, diameter, inches	7.87
stroke, inches	11.81
Sanitary pump (1), single-cylinder, diameter, inches	11.81
stroke, inches	11.81
Screw propellers, number of blades	4.0
diameter, feet	21.98
pitch, feet	22.15
Pumps, air (2), Blake, size	14.0 × 32.0 × 21.0
main feed (3), Weir, double, size	17.0 × 12.0 × 26.0
auxiliary feed (2), Vulcan, duplex, size	14.17 × 9.0 × 12.2
bilge (2), Vulcan, duplex, size	6.5 × 7.8 × 13.78
sanitary (2), Vulcan, duplex, size	11.0 × 7.8 × 13.78
auxiliary (1), Blake, size	9.0 × 20.0 × 13.0
evaporator (2), size	3.0 × 2.0 × 3.0
ballast (3), Vulcan, duplex, size	11.8 × 11.0 × 11.8
auxiliary condenser (1), crank and fly-wheel,	... 7.0 × 7.0
fresh water (2), size	7.0 × 6.0 × 7.60
(1), size	7.0 × 6.7 × 7.87
hydraulic (1), size	3.3 × 1.8 × 3.9
ice machine (1), fly-wheel, size	... 6.0 × 8.0
Fans, for Howden draft (2), 8.66 × 16.5 × 7.87, diameter, inches	134.0
E.R. ventilation (2), 7.5 × 6.3, diameter, inches	94.5
F.R. ventilation (2), 6.3 × 11 × 7, diameter, inches	118.0

Condensers, main (2), diameter, feet, 7.38, cooling surface, square feet	24,220
tubes, diameter, inch	⅝
thickness, inch	0.055
length, feet	15.8
number in each	3,824
auxiliary (1), diameter, feet, 4.43, cooling surface, square feet	250
auxiliary, tubes, diameter, inch	⅝
thickness, inch	0.051
length, feet	8.8
number in each	1,496
Ash ejectors (4), inches	6

Dynamo engines (6), each of 800 ampères and 110 volts, with cylinders 11.8 by 19.7 by 10.8.

Two ice machines of 90,000 W.E. } For driving the larger
One aux. ice machine of 7,000 W.E. } there is an engine, 9.8 by 15.75 by 11.8, and for the smaller one a single-cylinder engine, 5.9 by 5.5.

Of the nine boilers, eight are double- and one single-ended, four of the double-ended ones being in the after boiler room, and the remainder in the forward one. The double-ended ones are worked on the Howden system of forced draft, and the single-ended one with natural draft only. All work with the same pressure, 213 pounds.

Boilers, D. E., diameter, feet	16.08
length, feet	20.25
furnaces, number	6.0
diameter, inches	45.27
grate surface, each, square feet	127.88
heating surface, each, square feet	5,398.0
S. E. diameter, feet	15.25
length, feet	11.0
grate surface, square feet	63.94
heating surface, square feet	2,110.0
grate surface, total, square feet	1,087.0
heating surface, total, square feet	45,292.0
Smoke pipes, number	2.0
height above grate, feet	126.6
length, fore-and-aft, feet	15.66
breadth, feet	11.68
Weight of one D. E. boiler, without water, tons	98.5
S. E. boiler without water, tons	52.2

—Translated from "Zeitschrift des Vereines deutscher Ingenièure."

Empress of Britain and Empress of Ireland—Trials of.—A general description of these vessels was given in pages 334-336.

When loaded to their mean service draught—viz., 27 feet 6 inches—the vessels were required to carry a load of 6,500 tons, including stores. As a matter of fact, it was found possible to carry 400 tons more. As regards speed, the ships were guaranteed by the builders to steam 19½ knots on the mean service draught. The *Empress of Britain* steamed 19.78 knots on her measured-mile trials, with the engines indicating 18,750 horsepower, while the *Empress of Ireland* on the same trial steamed 20,002 knots with the engines developing 19,200 indicated horsepower. On the long-distance coal-consumption trials at 18 knots' speed the results were as follows:

	Empress of Britain.	*Empress of Ireland.*
Horsepower	13,450.0	13,000.0
Speed, knots	18.6	18.0
Coal consumption, pounds	1.33	1.37

These results have been fully maintained since the vessels went on service. They have succeeded in lowering the Atlantic record to Canada, and their popularity has justified the enterprize of the Canadian Pacific Company, and rewarded them for the great care they displayed in the details of the design.

La Provence.—This new speedy steamer, constructed by the Society of Shipbuilders of St. Nazaire, at Penhoet, France, represents the latest type of French mercantile steamer, and is the third and last of three steamships to be built for postal service between Havre and New York, in accordance with the agreement entered into July 6, 1898, between the State of France and the General Trans-Atlantic Co. The other two being *La Lorraine* and *La Savoie*.

The dimensions of *La Provence* are: Length over all, 627 feet; length between perpendiculars, 597 feet; beam, 65 feet; depth amidships, 45 feet; displacement, 19,100 tons; normal draught, 26.75 feet; horsepower, 30,000; speed, trial trips, 22.7 knots.

There are six steel decks connected by elevators. The passenger capacity is 1,604 and the personnel of the ship comprises 446 persons. Coal bunkers hold 35,000 tons.

Propulsion is by twin screws operated by triple-expansion four-cylinder engines of an aggregate of 30,000 horsepower, the trial speed having been somewhat less than twenty-three knots. Steam is supplied by twenty-one cylindrical boilers working at a pressure of 206 pounds per square inch, and fitted with the Howden system of forced draft. The coal consumption per day is given as 450 tons, while the bunkers have a total capacity of 4,400 tons.

The cylinder diameters are respectively 120, 194 and two of 242 centimeters, with a stroke of 170 centimeters. These figures correspond respectively to cylinder diameters of 47¼, 76½ and 95¾ inches, the stroke being 67 inches. The boilers have mean diameters of 17 feet 1 inch and 16 feet 1 inch in the two sizes, with a length of 10 feet 10 inches in each case. The aggregate grate surface is 1,570 square feet with heating surface 58,172 square feet, the ratio of the one to the other being 37 to 1.

On trial trip the speed reached was 22.7 knots, with an average of 22.3, at a draught of 26 feet 11 inches. The average coal consumption per indicated horsepower per hour figured out at about 600 grams, which is 1.32 pounds, a remarkably good performance. The maximum power developed was 30,625 indicated horsepower. The revolutions were 87 per minute.

OBITUARY.

The Society regrets to announce the death of two of its members since the May issue of the JOURNAL.

H. S. DAVIS, Associate, connected with the Wm. Cramp and Sons Ship and Engine Building Company, died May 10, 1906, in Philadelphia, Pa.

Naval Constructor JOSEPH JANVIER WOODWARD, U. S. N., died, near Field, British Columbia, July 6, 1906.

Naval Constructor Woodward was born in Philadelphia, Pa., October 2, 1860.

He was a member of the class of 1881 at the Naval Academy, and later was sent to France for a course in naval architecture.

He had duty at the Navy Yard, New York, the Bureau of Construction and Repair, the Newport News Shipbuilding and Dry Dock Company, and at the time of his death was a member of the Board of Inspection and Survey.

Constructor Woodward ranked high in his profession, and in his death the Naval Service has lost one of its most useful officers.

BOOKS RECEIVED.

THE NAVAL POCKET BOOK FOR 1906.—By GEOFFREY S. LAIRD, Clowes, London; W. THACKER & Co., 2 Creed Lane, E. C. Price 7s. 6d., net.

This invaluable little book, founded by Sir W. Laird Clowes, is now edited by his son. The arrangement of the book is unchanged, but the Navies of a number of countries, particularly Brazil, Holland, Japan and Russia, have been revised in considerable detail, and about twenty new plans of ships have been added to the illustrations.

A summary of the contents is as follows: Calendar and Naval Remembrances, 1906; Classification of Ships; Comparative Summary of the Fighting Fleets; Classified List of the Navies of the World; Guns and Small Arms; Notes on Torpedoes; Naval Personnel and Comparative Rank; Retiring Ages in European Navies; List and Particulars of Dry Docks; Miscellaneous Tables; Trial-Trip Tables; Plans of Ships.

There are 965 pages of the most useful data for reference, and the little volume should be in the hands of every Naval officer.

ASSOCIATION NOTES.

At a meeting of the Council held on Monday, August 6, 1906, at the Navy Department, Commander B. C. Bryan, U. S. Navy, was elected President of the Society, to fill the vacancy caused by the detachment from duty in the Department and orders to Boston of Commander A. B. Canaga, U. S. N.

Lieutenant Commander Harold P. Norton, U. S. Navy, was elected a member of the Council to fill the vacancy created by Commander Bryan's election to the Presidency of the Society.

The Council authorized the Secretary-Treasurer to purchase for the Society five (5) one thousand dollar bonds of the Washington Railway and Electric Co.

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